

COMPUTER AIDED PERFORMANCE STUDY & COST ANALYSIS OF A DOUBLE-STAGE-VAPOUR ABSORPTION SYSTEM

By

ARUN MEHTA



DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

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A Thesis Submitted
In Partial Fulfilment of the Requirements
for the Degree of

MASTER OF TECHNOLOGY

By

ARUN MEHTA

to the

DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

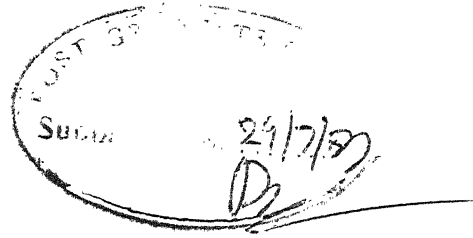
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CERTIFICATE



This is to certify that this work on Computer Aided Performance Study and Cost Analysis of a Double-Stage-Vapour-Absorption-System has been carried out under my supervision and it has not been submitted elsewhere for a degree.

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Department of Mechanical Engineering,
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July, 1987

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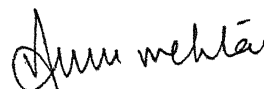
With immense pleasure, I express my deep sense of gratitude and regards to Professor H.C. Agrawal, for inspiring me to take up this problem and providing his inspiring guidance and constructive criticism to overcome all difficulties encountered during this period. His intense interest and unflinching assistance have been the main factors responsible for the completion of this work in time.

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Arun Mehta

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NOMENCLATURE

A	Surface area	[m ²]
C _p	Specific heat	[kJ/kg. K]
d	Diameter of tube	[m]
f	Friction factor	
g	Acceleration due to gravity	[m/s ²]
H	Enthalpy	[kJ/kg]
h	Heat transfer coefficient	[W/m ² .K]
K	Thermal conductivity	[W/m.K]
\dot{m}	Mass flow rate	[kg/s]
N	Number of tubes per pass	
P	Pressure	[k Pa]
Q	Volume flow rate	[m ³ /s]
q	Heat load	[kW]
R	Thermal resistance	[m ² .K/W]
t	Temperature	[°C]
t _f	Temperature	[°F]
TR	Refrigerant temperature	[°C]
TS	Solution temperature	[°C]
U	Overall heat transfer coefficient	[W/m ² .K]

V	Velocity	[m/s]
X	Lithium bromide concentration %.	[kg/kg of sol.]

Subscripts

a	Referred to absorber
c	Referred to condenser condition
e	Referred to evaporator condition
ff	Fouling condition
g1	Referred to first stage generator
g2	Referred to second stage generator
i	Referred to inlet condition
m	Mean
o	Referred to outlet condition
t	Tube side
w	Water side
1,2,3,,,7	State points referred to Fig. (1.3) for single-stage
1,2,3,..18	Stage points referred to Fig. (1.4) for double-stage
I,II	First and second heat exchanger

Greek Symbols :

μ	Dynamic Viscosity	[Pa.s]
ϵ	Absolute roughness	[mm]
ρ	Density of fluid	[kg/m ³]
ν	Specific volume	[m ³ /kg]

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ABSTRACT

The escalating cost of energy and the fast depleting sources of conventional-energy have sounded an alarm to the world, urging us to be meticulous in using any form of energy, at our disposal. Using waste heat energy from various industrial processes and the use of non-conventional energy sources, such as : solar, geothermal, bio-gas, wind, tidal etc., are vitally important propositions in this context. A double-stage-vapour-absorption-system to produce refrigeration, attempts to fulfil the efficient use of the above energy resources.

Keeping the above fact in mind, the present work is devoted to the analysis of a double-stage-vapour-absorption-system. The complete performance-study of the system is carried out to determine the variation of C.O.P. (the index of performance of the system) with various operating parameters. The optimum values of these parameters are also evaluated for maximum C.O.P. This C.O.P. is compared with the C.O.P. of the corresponding single-stage-vapour-absorption-system.

The most conventional method of producing refrigeration is by using vapour-compression-system, which uses the valuable mechanical energy for its operation. Therefore, is the major

competitor to the proposed system, presented in the thesis. Hence, a comparative cost-analysis is also undertaken in the present study to evaluate the relative economics of operation for the two systems.

A digital computer DEC-1090 has been used to help in carrying out the analysis. Required computer programmes have been developed to perform this analysis, which are made very general to encompass a class of similar problems. Results have been discussed and the following conclusions drawn :

1. The C.O.P. obtained in a double stage-system is quite high (1.785 maximum) as compared to single stage (0.84 maximum). However, very high input source-temperature is required for system's operation.
2. This system has very high values of C.O.P., almost approaching the ideal value at high condenser temperatures. The practicability of using high condenser-temperatures obviates the necessity of using a cooling tower along with the condenser and an air cooled condenser may be used for the purpose. This helps in a reduction in total cost of the system
3. A double-stage-vapour-absorption-system is more economical than a vapour-compression-system in terms of the annual running cost as it does not require compressor for its operation, which consumes the operation of a vapour-

CHAPTER-I

VAPOUR ABSORPTION AND COMPRESSION SYSTEMS

1.1 INTRODUCTION :

Vapour compression system is invariably used in most of the refrigeration and air conditioning industries. The outline of such a system along with all its basic components ^{is} ~~are~~ shown in Fig. 1.1. However, it uses the costly mechanical energy in the form of compressor work and the ever-rising costs of this energy due to continuous depleting energy resources have demanded that this valuable energy should be conserved, as much and as far as possible. There are broadly two ways which can assist in conserving the conventional energy :

- (i) using non-conventional energy sources like solar, tidal, bio-gas, geothermal and wind energy, economically and
- (ii) using waste energy to recover some heat out of it, which otherwise would be wasted. Waste heat in the form of air and hot water from industries such as steel, chemical, textile, foundaries etc. can be considered as practical examples.

Vapour absorption system, which can work using either of the above two alternative energy sources or their

combination, is one of the very old methods to produce cold. Single-stage-vapour-absorption-system was first used in U.S.A. during the civil war after the supply of natural ice had been cut off from North. Many improvements have been incorporated in the system over the years. The skeleton diagram of such an improved version of single-stage-vapour-absorption system is shown in Fig. 1.2. Since, the thesis work is devoted to the vapour-absorption system, we describe below the various components of this system:

(i) Generator

It is basically a heating vessel, which receives heat from the heat source and transfers it to the solution flowing inside the tubes. The source of heat can be steam, solar-heat, bio-gas, hot-water, hot-air etc.

(ii) Condenser

It is a shell and tube type heat exchanger located after the generator. Cooling water to cool refrigerant flows inside the tubes, whereas, refrigerant condenses in the shell.

(iii) Absorber

It is a heat-rejecting vessel. The strong absorbent solution in the absorber is sprayed to absorb the low pressure and temperature refrigerant vapour, coming out of the evaporator. Heat of absorption, which is generated due to exothermic

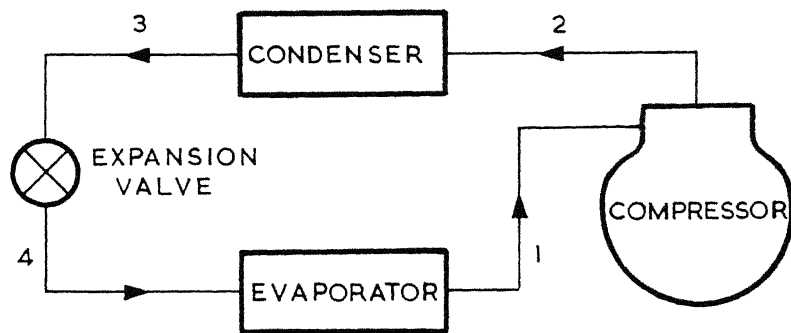


FIG.1.1 FLOW DIAGRAM OF VAPOUR COMPRESSION SYSTEM.

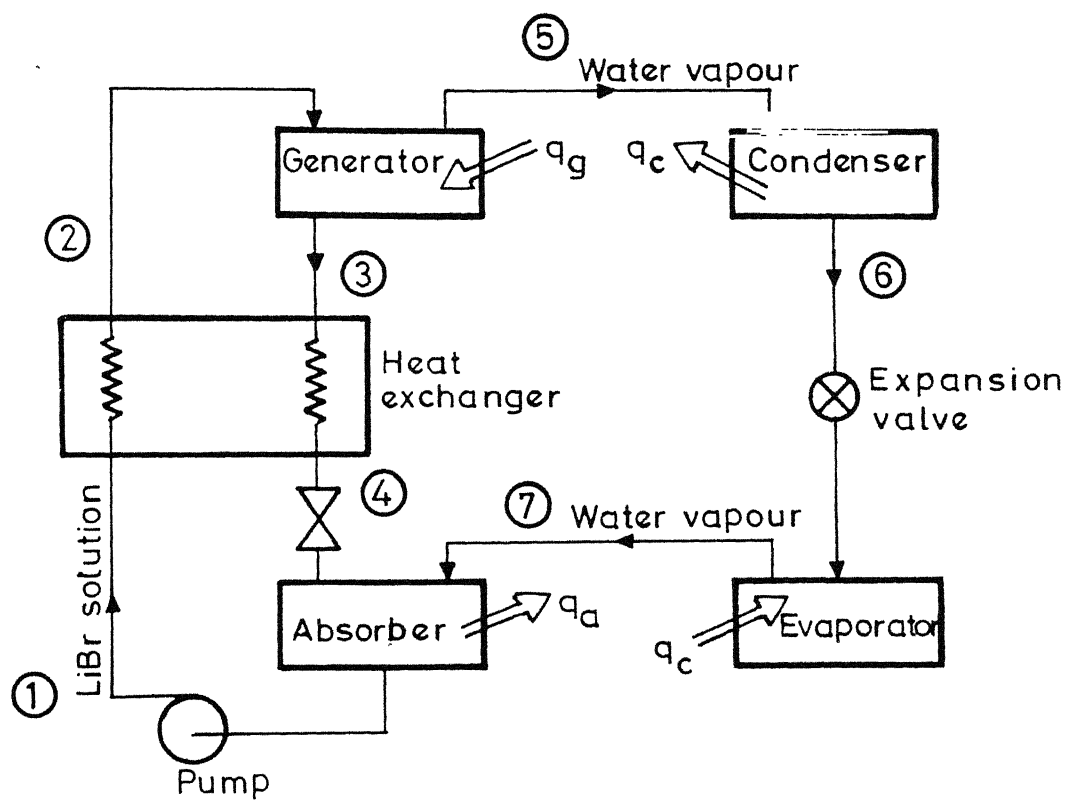


FIG. 1.2 ONE-STAGE ABSORPTION SYSTEM WITH HEAT EXCHANGER .

reaction, is rejected to the atmosphere, through the cooling water which flows inside the tubes.

(iv) Evaporator

Evaporator, sometimes also known as the Chiller, gives the desired cooling effect. Refrigerant flows inside the tubes, whereas liquid to be cooled flows over the tubes. Like condenser, it also has a shell and tube type construction.

(v) Solution-Heat-Exchanger

Solution-heat-exchanger is of shell and tube type construction. It heats the weak solution on its path from the absorber to the generator by extracting heat from the relatively hotter strong solution returning from the generator. This reduces the amount of heat to be added to the solution in the generator.

(vi) Expansion-Device

Expansion-device i.e. expansion valve is used for the reduction of pressure and temperature of the refrigerant liquid, while flowing from the condenser to evaporator. The valve used is generally an orifice or other fixed restriction $\sqrt{11}$ type.

(vii) Solution Pump

It is generally electrical motor-driven centrifugal-pump of hermetic design, which circulates the solution in

The index of performance of a refrigeration system is represented by a dimensionless parameter termed as 'Coefficient of Performance', abbreviated as C.O.P.

For a vapour compression system, C.O.P. is defined as :

$$\text{C.O.P.} = \frac{\text{Useful refrigerating effect}}{\text{Net work done by the compressor}}$$

For a vapour absorption system, on the other hand, the C.O.P. is expressed as :

$$\text{C.O.P.} = \frac{\text{Useful refrigerating effect}}{\text{Rate of heat addition in the generator}}$$

An absorption cycle has an appreciably lower value of C.O.P., than that of a compression cycle (0.8 versus 3.5). Vapour compression cycle is ^awork operated cycle (mechanical work in compressor), whereas, absorption cycle is ^aheat operated cycle (heat addition in generator) and energy in the form of work is normally much more valuable and expensive than energy in the form of heat. It is, therefore, obvious that, the two systems cannot be measured on the same scale. The vapour compression system pushed absorption system ~~as to~~ ^{to} ~~obsolete~~ because of its being more reliable in operation and having ^abetter C.O.P. However, with the advent of worldwide concern over energy conservation, the vapour absorption system has again gained importance as it saves ~~it~~ of

conventional, valuable, mechanical energy. This system has been found quite attractive in the space-conditioning applications, producing chilled water for various application and for preserving perishable commodities.

Two types of vapour absorption systems can be practised:

- (i) using Lithium Bromide as the absorbent and water as the refrigerant and
- (ii) using water as the absorbent and Ammonia as the refrigerant.

The combination adopted in the second type is called aqua-ammonia and was used in absorption systems, years before the $\text{LiBr-H}_2\text{O}$ combination became popular, which has now almost completely replaced the aqua-ammonia system due to some of its following distinct advantages :

- (i) this combination yields higher C.O.P. at most operating temperatures.
- (ii) it operates at lower system pressures, thereby, requiring low pumping power.
- (iii) it can be used in commercial and residential buildings without any restriction from safety consideration.
- (iv) lower first cost due to the absence of parts like rectifier and analyzer.

Various improvements and refinements in conventional old Single-stage-absorption-system have been introduced from time to time to make vapour absorption system more efficient. It includes the reducing the generator temperature for system operation, increasing the allowable cooling water temperature and increasing the coefficient of performance, which, in turn, improves the efficiency of the system and conserves the energy.

The practicability of increasing the cooling water temperature is of special importance, because a sufficient reduction in the cost of the system may be achieved by using an air-cooled heat exchanger for heat rejection from the sinks of the absorption system and the requirement of a cooling tower may be eliminated.

1.2 LITERATURE-SURVEY

According to available literature tests for proving feasibility of solar air conditioning appeared in 1960 [1]. Design of one ton solar operated LiBr-H₂O air conditioning system was presented at National Solar Energy Convection at Bhavnagar [2] and performance prediction of LiBr-H₂O absorption air conditioning system, utilizing the solar energy was put forward at International Solar Energy Congress at New Delhi [3].

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Design and optimization of an absorption cycle with $\text{LiBr-H}_2\text{O}$ and $\text{H}_2\text{O-NH}_3$ combination was studied by Alizadeh.et.al. [4_7]. This study shows that, in general, for fixed initial conditions and given system-refrigeration-capacity, higher generator temperature causes higher cooling ratio with smaller heat exchange surfaces and, consequently, lower cost. A comparison of the two cycles also indicates that $\text{LiBr-H}_2\text{O}$ system is much simpler than $\text{NH}_3\text{-H}_2\text{O}$ system and operates at a higher cooling ratio and for smaller heat exchange surfaces for the same conditions.

Design aspects of an air-cooled Lithium-Bromide-Water absorption air conditioning system have been discussed by W.W.S. Charters et.al. [5_7], which include information on relative circulation rate of the solution and heat-exchanger effectiveness. Details are also provided for alternative design aspects in case crystallization of the solution occurs.

An economic evaluation of the use of Bio-gas to run a $\text{LiBr-H}_2\text{O}$ absorption system has been carried out by Siddiqui et.al. [6_7]. Coefficient of Performance of the system has been studied for various values of generator-temperature, pressure, evaporator-temperature, precooler and preheater effectiveness.

A dynamic model of solar air conditioning, including the solar collector and cooling tower, in terms of design parameters, is described by S.L. Grassie [7]. It has the special feature that it provides a refrigerant storage tank associated with the generator to stabilize the flow.

All the work listed above relates to single-stage-absorption-system. Due to escalating energy cost, in recent times, more attention, however, is being paid towards the energy conservation and its effective utilization. This has necessitated the study of a more efficient and improved system. Double-stage-vapour-absorption-system is such an improvement, which is supposed to work on higher C.O.P. than a single stage with certain other advantages like reduced condenser load, material saving in the tubes etc.

One of the recent works on the study of a double-stage-vapour-absorption-system is done by Kaushik and Kumar in the year 1985 [8], in which they have considered the refrigerant-absorbent combination of $\text{NH}_3\text{-H}_2\text{O}$ and $\text{NH}_3\text{-LiNO}_3$. The system consists of coupling of two conventional absorption cycles, so that the first stage evaporator produces cooling water to be circulated in the absorber of the second stage. The effect of operating variables, such as generator-temperature, evaporator-temperature and condenser-temperature on the C.O.P., heat transfer rates and relative circulation rates, has been

studied. Reference has been made by Stoecker and Jones [9_7] to the design of a double-stage-vapour-absorption-system proposed by the Trane Company of U.S.A.. LiBr-H₂O combination has been used as the working fluid for the system. A circuit diagram is presented (Fig. 1.3). C.O.P., of the order of 1.0 to 1.4 has been reported for such a system.

1.3 PRESENT WORK

The present work is an extension of the design concept for the double-stage-vapour-absorption-system, enunciated by the Trane Company [9_7]. The same working fluid i.e. LiBr-H₂O combination is used. An improved design of the system has been proposed. The detailed circuit diagram for the same is shown in (Fig. 1.4). This design has the following merits in comparison with the one, given by the Trane Company.

- (i) The solution entering the generator-II is heated by the steam from generator-I, which drives off some water vapour from the LiBr-H₂O solution. Hence, the solution leaving generator-II, is a two-phase mixture. This two-phase mixture is then throttled to the condenser pressure in throttling-valve-I. Since, the throttling valves are not designed to handle two phase mixture, this operation is inefficient. On the other hand, in the improved system, presented in Fig.(1.4), no throttling of LiBr-H₂O solution is necessary, as generator II is already maintained at condenser pressure.

- (ii) In the system, developed by the Trane Company, generators-I and II are put in the same vessel. The thermodynamic properties of the $\text{LiBr-H}_2\text{O}$ solution are such that, they impose a restriction on the heat exchanger-II. The level of cooling to be achieved in this heat exchanger for the $\text{LiBr-H}_2\text{O}$ solution, entering the generator-II should be sufficient to accept heat from the steam, given off from generator-I. But in the system, proposed in the present work Fig. 1.4), generator-II is uncoupled from generator-I and, therefore, no restriction on the cooling capacity of the heat exchanger need be imposed.

The performance characteristics of the improved system are studied by the use of numerical modelling of the system which is based on the corresponding mass, energy and material balance equations of each component of the system. Effect of various component's temperature viz generator, absorber, condenser and evaporator on the performance of the system has been studied. Optimization of generator-I temperature for maximum C.O.P. along with minimum generator-temperature required to carry out system operation have been determined. Effect of heat-exchanger effectiveness on the system performance has also been studied.

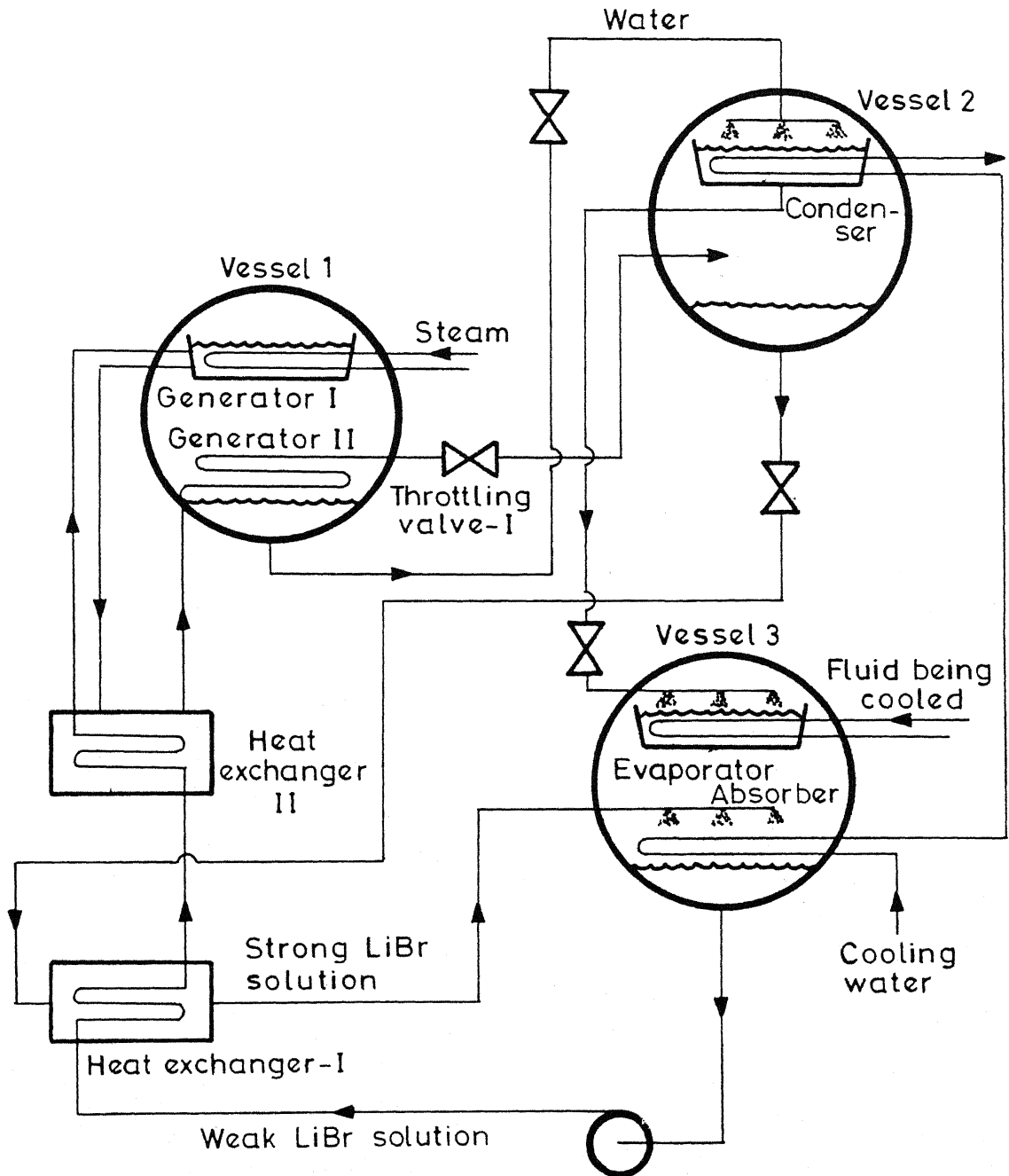


FIG.1.3 DOUBLE-EFFECT ABSORPTION UNIT (THE TRANE COMPANY).

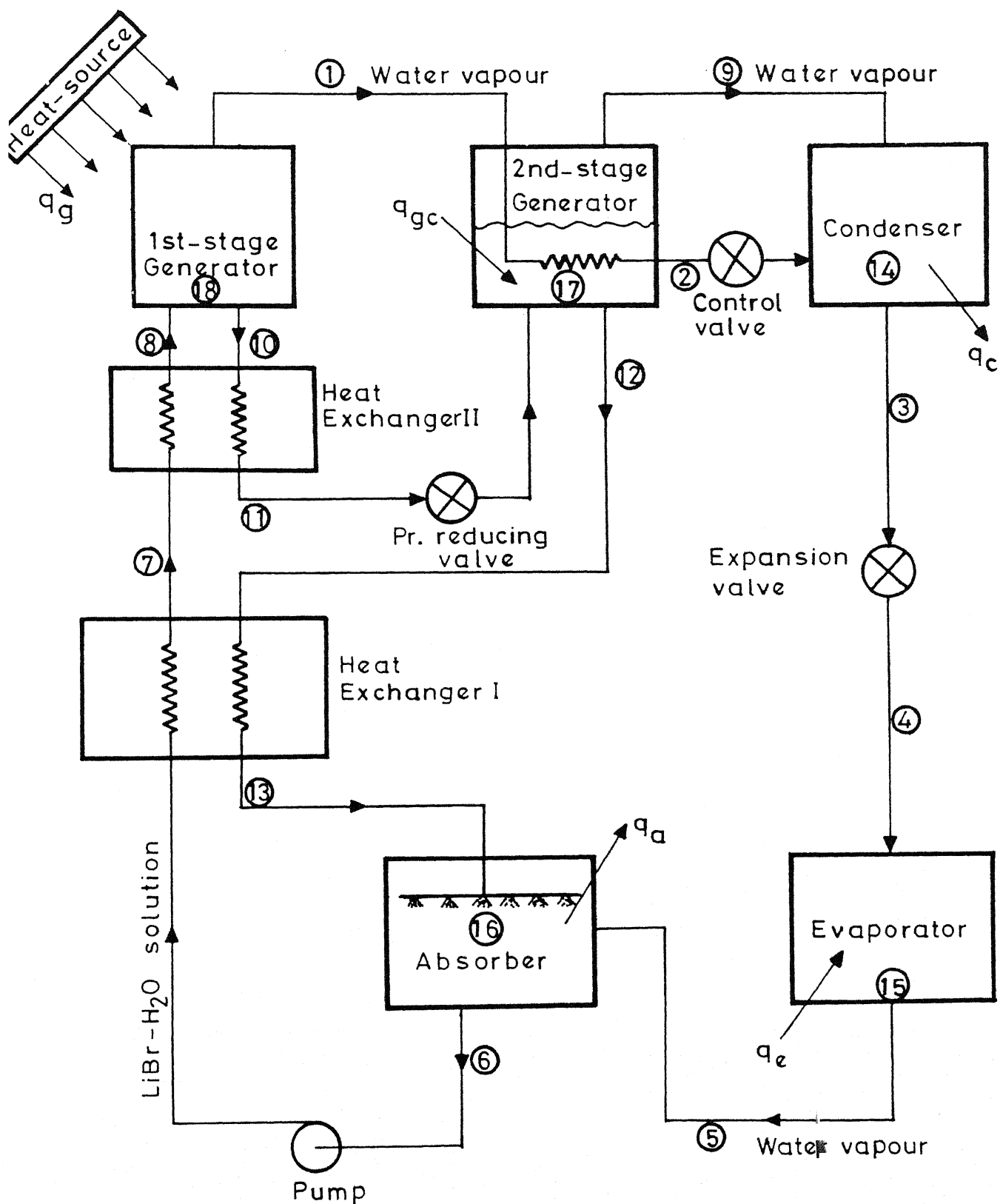


FIG.1.4 SCHEMATIC DIAGRAM OF A DOUBLE STAGE VAPOUR ABSORPTION SYSTEM (PRESENT WORK)

Comparison of the performance of this system is carried out with that of corresponding single-stage-vapour-absorption-system.

General computer programmes in FORTRAN language have been developed for both the single-stage and double-stage systems to evaluate the operational parameters (pressure, temperature, concentration, heat flow rates, mass flow rates of solution, refrigerant etc.), where-ever needed.

To emphasize the economic advantages of the double-stage LiBr-H₂O vapour absorption system, it's yearly operating cost has been compared with that of the vapour-compression-system of the same capacity. Suitable computer programme has been developed for this purpose. Operating cost includes the cost of power used in pumping the LiBr-H₂O solution and cooling water and also in raising the pressure level of the LiBr-H₂O solution from the absorber to the generator-I (Fig. 1.4).

For drawing graphs, 'LOTUS' subroutine in IBM personal computer has been used.

CHAPTER-II

SINGLE-STAGE-VAPOUR-ABSORPTION SYSTEM

2.1 DESCRIPTION OF THE SYSTEM:

The basic cycle of single-stage LiBr-H₂O vapour-absorption-system (Fig.1.2) consists of four temperature levels and two pressure level-zones.

Temperature level zones are :

- (i) Generator (T_g)
- (ii) Condenser (T_c)
- (iii) Absorber (T_a)
- (iv) Evaporator (T_e)

Pressure level zones are :

- (i) the low pressure prevailing in the evaporator and absorber.
- (ii) the high pressure prevailing in the generator and condenser.

In the system operation, the weak solution of LiBr-H₂O is pumped from the absorber to generator through a solution heat exchanger. In the generator, the solution is heated at high temperature (60-90°C), so that, some of the refrigerant i.e. water is vaporised from the LiBr-H₂O solution. The

which is strong solution (due to lower

the heat exchanger. In the heat exchanger, this solution transfers some of its heat to the weak solution, pumped from the absorber. A throttling valve is also provided to develop a pressure drop between generator and absorber ($p_g > p_a$).

The refrigerant vapour leaving the generator, passes to the condenser and is condensed, rejecting the heat of condensation to the circulating cooling water. The refrigerant-liquid in the condenser passes through the expansion valve and evaporates in the evaporator at very low pressure and temperature, extracting latent heat of vaporisation from the surroundings. It causes the cooling effect in the surrounding space. This vapour is then dissolved in the strong solution, returning back from the generator to absorber. This process produces heat of solution, which is carried away by the cooling water. After absorption, the weak solution is again pumped back to the generator. This completes the cycle of operation.

2.2 THERMODYNAMIC ANALYSIS OF THE SYSTEM :

A detailed thermodynamic analysis of the single-stage-absorption cycle has been carried out to evaluate the C.O.P. of the system. The following assumptions have been made :

- (i) the system is in a steady-state.
- (ii) the properties of the solution leaving a component are representative of the corresponding solution-

- (iii) pressure loss in the heat exchanger is neglected.
- (iv) minimum terminal temperature difference is 5°C in the heat exchanging components of the system.
- (v) pressures in the evaporator and condenser are equal to the vapour pressure of the refrigerant i.e. water, at the corresponding evaporator and condenser temperatures.

Thermodynamic Analysis :

- (1) The absorber-temperature and the condenser-temperature are fixed by the available cooling water temperature. The requirements of the conditioned space fixes the evaporator-temperature. Heat exchanger effectiveness is assumed to be 0.9 which is in the commonly used range of its value 10. The available heat-source-temperature determines the generator-temperature.
- (2) The generator and the condenser, being directly connected to each other, exhibit the same pressure in them, i.e. $p_g = p_c$. For similar reason, pressures in the evaporator and the absorber are also equal i.e. $p_e = p_a$.

The condenser pressure ' p_c ' as a function of condenser temperature ' T_c ' is given in expression (A-2) of Appendix-1 which describes the state equations

for aqueous LiBr solution. Since ' T_c ' is already known, ' P_c ' can be evaluated using (A-2). Similarly, from the prior knowledge of ' T_e ', ' P_e ' can be determined.

- (3) Using the already known values of absorber pressure and temperature, the concentration of LiBr-H₂O solution, leaving the absorber ' X_1 ' can be read from the graphs given in Appendix-2. Similarly, ' X_3 ', the concentration of the LiBr-H₂O solution, leaving the generator is obtained from the knowledge of generator pressure and temperature. Further as there is no change of concentration in the heat exchanger, we must have $X_1 = X_2$ and $X_3 = X_4$.

- (4) The conservation of mass for the working fluid in the generator is given by ;

$$\begin{aligned} &\text{rate of inflow of total mass to the generator } (\dot{m}_2) \\ &= \text{rate of outflow of total mass from the generator} \\ &\quad (\dot{m}_3 + \dot{m}_5) \end{aligned}$$

or,

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_5 \quad (2.1)$$

Similarly, conservation of LiBr in the generator requires,

$$\dot{m}_2 X_2 = \dot{m}_3 X_3 \quad (2.2)$$

At this stage, a term, called the Absorption-Recirculation-Ratio (ARR) is defined. 'ARR' is the mass of solution, pumped to the generator per unit mass of water vapour, supplied to the condenser. The 'ARR' is represented in terms of mathematical expression as :

$$ARR = \frac{\dot{m}_2}{\dot{m}_5} \quad (2.3)$$

Using (2.1) and (2.2), the expression for ARR reduces to,

$$ARR = \frac{X_3}{X_3 - X_2} \quad (2.4)$$

(5) The conservation of total mass in the absorber gives,

$$\dot{m}_1 = \dot{m}_4 + \dot{m}_7 \quad (2.5)$$

Also, the conservation of LiBr in the absorber yields,

$$\dot{m}_1 X_1 = \dot{m}_4 X_4 \quad (2.6)$$

Another term, called the Generator-Recirculation-Ratio (GRR) is defined. 'GRR' is the ratio of mass of LiBr-H₂O solution, leaving the generator to the mass of water-vapour driven off from the generator. Mathematically, 'GRR' is given by,

$$GRR = \frac{\dot{m}_3}{\dot{m}_5} = \frac{\dot{m}_4}{\dot{m}_7} \quad (2.7)$$

Using relations (2.5) and (2.6), the expression (2.7) reduces to ,

$$GRR = \frac{X_1}{X_4 - X_1} \quad (2.8)$$

- (6) The enthalpies of the fluid at various state points are to be determined in order to determine the heat loads on the various components in the system. In this context, enthalpies for LiBr-H₂O solution are obtained from the graph in Appendix-3 and the steam table [9_7] is used to get the enthalpy values for the refrigerant. Adopting this method, enthalpies at state points '1', '3', '5', '6' and '7' (Fig. 1.2) can be directly obtained. The corresponding enthalpies at state points '4', '2' are determined as following using the enthalpy values obtained above :

(a) Determination of H₄:

Heat-exchanger effectiveness is defined as :

$$\eta = \frac{\text{Enthalpy drop in the heat exchanger}}{\text{Maximum possible enthalpy drop in the same}}$$

or,

$$\eta = \frac{H_3 - H_4}{H_3 - H_1} \quad (2.9)$$

In the above expression, all the parameters are known, except H₄. Hence, H₄ can be evaluated by the relation :

$$H_4 = H_3 - \eta (H_3 - H_1) \quad (2.10)$$

(b) Determination of H_2 :

Energy-balance in the heat exchanger gives,

$$\dot{m}_3(H_3-H_4) = \dot{m}_1 (H_1-H_2) \quad (2.11)$$

Since \dot{m}_3 and \dot{m}_1 are not known exclusively, both sides of expression (2.11) are divided by \dot{m}_5 . With this modification, expression (2.11) reduces to ,

$$\frac{\dot{m}_3}{\dot{m}_5} (H_3-H_4) = \frac{\dot{m}_1}{\dot{m}_5} (H_1-H_2) \quad (2.12)$$

Using expression (2.3) and (2.7) in (2.12), we get,

$$GRR (H_3-H_4) = ARR (H_2-H_1)$$

or,

$$H_2 = \frac{GRR}{ARR} (H_3-H_4) + H_1 \quad (2.13)$$

- (7) The heat transfer rates to and from the various system components can now be calculated by performing energy balance for each component :

- (a) The heat flow rate in the evaporator :

$$q_e = H_7-H_6 \quad (2.14)$$

- (b) Heat, to be rejected in the condenser :

$$q_c = H_5-H_6 \quad (2.15)$$

(c) Heat, to be supplied in the generator :

$$q_g = \dot{m}_5 H_5 + \dot{m}_3 H_3 - \dot{m}_2 H_2 \quad (2.16)$$

Assuming \dot{m}_5 as unity and dividing both sides of the expression (2.16) by \dot{m}_5 , we obtain,

$$q_g = H_5 + GRR \times H_3 - ARR \times H_2 \quad (2.17)$$

(d) Heat, to be rejected by the absorber :

$$q_a = \dot{m}_4 H_4 + \dot{m}_7 H_7 - \dot{m}_1 H_1 \quad (2.18)$$

Again, with necessary simplifications, we get,

$$q_a = GRR \times H_4 + H_7 - ARR \times H_1 \quad (2.19)$$

(8) For the total balance of energy in the system,

$$\text{Total heat addition} = \text{Total heat rejection}$$

or,

$$q_g + q_e = q_c + q_a \quad (2.20)$$

This condition is very much essential, to be satisfied.

The Coefficient of Performance of the system is

then given by

$$C.O.P. = \frac{q_e}{q_g} \quad (2.21)$$

The value of C.O.P., obtained by the above procedure is then compared with the C.O.P. of the ideal system to get an idea of how closely the system approaches the ideal cycle. The procedure to obtain C.O.P. of the ideal system is given below :

An ideal vapour-absorption-cycle can be considered as a combination of a power cycle and refrigeration cycle (Fig. 2.1). The power cycle receives energy in the form of heat q_g at an absolute temperature T_g , delivers some energy W in the form of work to the refrigeration cycle and rejects a quantity of energy q_a in the form of heat at a temperature T_a . The refrigeration cycle receives the work W which is pumped along with the heat q_e at the refrigerating temperature of T_e to a temperature T_c , where the quantity q_c is rejected.

The power-cycle, shown in Fig. (2.1), operates between the higher temperature limit T_g and the lower temperature limit T_a . Since, the analysis is carried out for an ideal cycle, the ideal thermal efficiency of this cycle is given by the Carnot's efficiency, which is represented as :

$$\begin{aligned} \eta_{\text{Power-cycle}} &= \frac{W}{q_g} \\ &= \frac{T_g - T_a}{T_g} \end{aligned} \quad (2.22)$$

Similarly, the refrigeration-cycle (Fig. 2.1), operates between sink temperature ' T_e ' and source temperature ' T_c '. For an ideal cycle, the C.O.P. is given by Carnot's C.O.P., represented as :

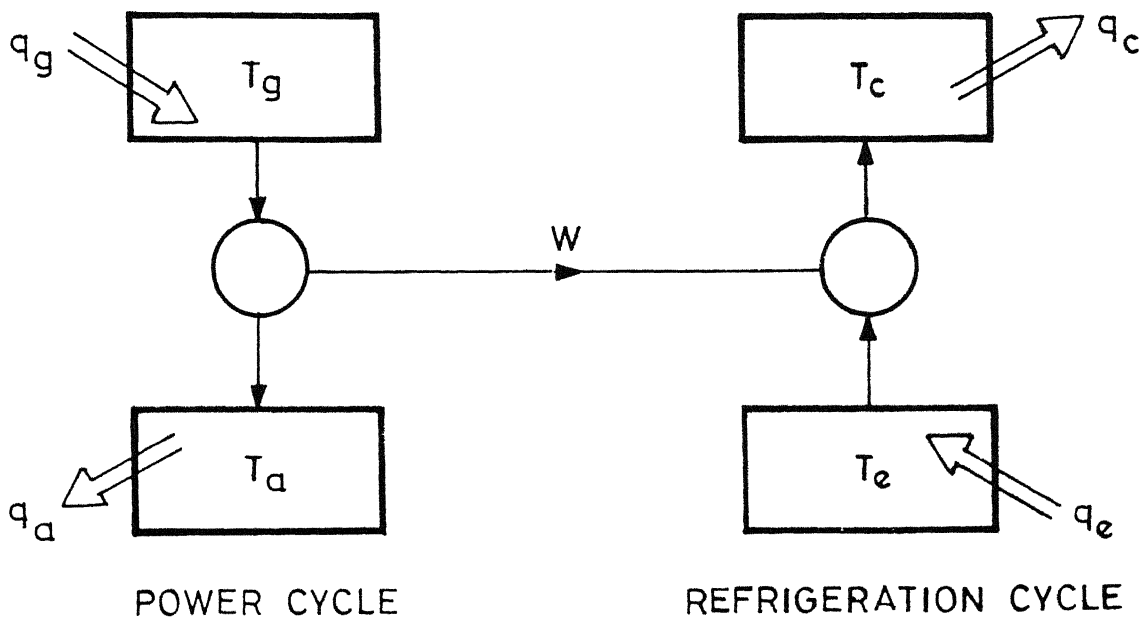


FIG. 2.1 HEAT-OPERATED REFRIGERATION CYCLE AS A COMBINATION OF A POWER CYCLE AND REFRIGERATION CYCLE.

$$\begin{aligned}
 \text{C.O.P.}_{\text{Ref}} &= \frac{q_e}{W} \\
 &= \frac{T_e}{T_c - T_e} \quad (2.23)
 \end{aligned}$$

The C.O.P. of the entire cycle is the multiple of $\eta_{\text{Power-cycle}}$ and $\text{C.O.P.}_{\text{Ref}}$ and is given by

$$\begin{aligned}
 \text{C.O.P.}_{\text{Ideal}} &= \eta_{\text{Power-cycle}} \times \text{C.O.P.}_{\text{Ref}} \\
 &= \frac{W}{q_g} \times \frac{q_e}{W} \\
 &= \frac{q_e}{q_g} \\
 &= \frac{T_e}{(T_c - T_e)} \cdot \frac{(T_g - T_a)}{T_g} \quad (2.24)
 \end{aligned}$$

2.3 COMPUTER PROGRAMME PACKAGE :

A computer programme has been developed to evaluate the C.O.P. of the single-stage-vapour-absorption-system, based on the thermodynamic analysis of the system described in section 2.2. Complete listing of this computer programme along with the description of the various input parameters and the execution commands on DECSYSTEM-1090 is given in Appendix-4.

The programme also includes the evaluation of pressure, temperature, concentration of LiBr-H₂O solution, enthalpy, mass flow rates of refrigerant and solution along with heat flow rates at various state points. The flow chart for the analysis is given in Fig. (2.2).

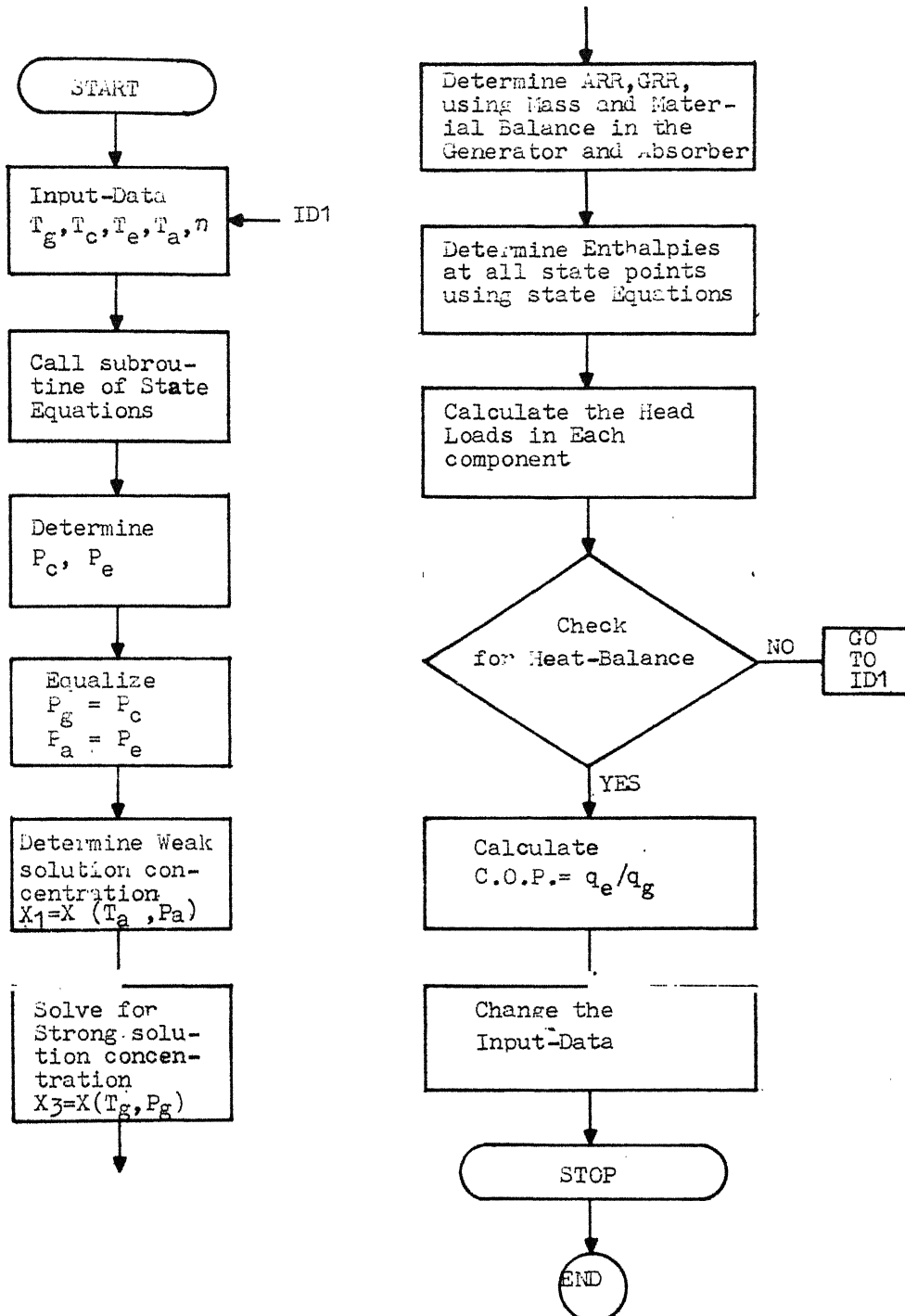


FIG.2.2 FLOW CHART FOR A SINGLE-STAGE-VAPOUR-ABSORPTION-SYSTEM.

CHAPTER-III

ANALYSIS AND COMPUTER MODELLING OF DOUBLE-STAGE-VAPOUR-ABSORPTION-SYSTEM

3.1 DESCRIPTION OF THE SYSTEM :

Fig. (1.4) shows schematic diagram of the proposed double-stage-vapour-absorption-system. It operates with five different temperature level zones and three distinct pressure zones, as following :

The temperature levels are those of

- (i) generator-I , (ii) generator-II, (iii) condenser,
- (iv) evaporator, (v) absorber

The pressure zones are :

- (i) the high pressure in the generator-I, determined by the saturation pressure corresponding to the refrigerant temperature in the generator-II. The zone indicates a pressure range of 50-70 KPa.
- (ii) the medium pressure in the condenser and generator-II, in the range of 3-5 KPa. It is the saturation pressure corresponding to the condenser-temperature.
- (iii) the low pressure prevailing in the evaporator and absorber, is determined by the saturation pressure corresponding to the evaporator-temperature. It ranges from 0.7 to 1 KPa.

Heat is supplied from the heat source (solar, bio-gas, hot-water, hot-air, industrial waste-heat etc.) to the generator-I for the system operation. This heats the weak solution of $\text{LiBr-H}_2\text{O}$, pumped from the absorber into the generator-I, through heat exchangers-I and II. A percentage of the refrigerant i.e. water evaporates out of this weak solution and gets condensed in the generator-II. The remaining $\text{LiBr-H}_2\text{O}$ solution in generator-I passes through heat exchanger-II and a pressure-reducing valve into the generator-II, where it receives latent heat from the refrigerant vapour evaporated out from generator-I. Some more percentage of refrigerant water from $\text{LiBr-H}_2\text{O}$ solution evaporates out and passes through the condenser. This evaporated water mixes with already condensed water coming out of the generator-II. Thus, the total amount of liquid refrigerant i.e. water leaving the condenser is the sum of the refrigerant originating from generators-I and II.

The refrigerant liquid from the condenser passes through an expansion valve, where its temperature and pressure lower down to the evaporator-temperature and pressure. In the evaporator, the refrigerant gets evaporated due to the latent heat of vaporisation, absorbed from the surroundings. This causes the cooling of space.

The cold vapours of the refrigerant, coming out of the evaporator are then dissolved in the strong solution of $\text{LiBr-H}_2\text{O}$,

collected in the absorber from the generator-II through the heat exchanger-I. A part of the heat of the strong solution of $\text{LiBr-H}_2\text{O}$ coming out of generator-II is transferred to the weak solution of $\text{LiBr-H}_2\text{O}$, pumped from the absorber. Also, when the strong solution of $\text{LiBr-H}_2\text{O}$ coming out of generator-II, mixes in the absorber with the low temperature and pressure water-vapour, coming out of the evaporator, it causes exothermic reaction and the heat so generated has to be rejected to the atmosphere by cooling the absorber. If the process of mixing is allowed to take place adiabatically (i.e. without rejecting the heat of absorption to the atmosphere), the temperature of the $\text{LiBr-H}_2\text{O}$ solution will rise in the absorber and, eventually, the absorption of low temperature and pressure water-vapour would not take place.

Once the strong $\text{LiBr-H}_2\text{O}$ solution mixes with the low temperature and pressure water-vapour in the absorber, the solution becomes weak and it is pumped out into the generator-I, as mentioned earlier. The pumping process elevates the pressure of the solution. Thus, the double-stage-vapour-absorption-cycle is completed.

3.2 THERMODYNAMIC ANALYSIS OF THE SYSTEM :

The thermodynamic analysis of the double-stage-vapour-absorption system is carried out to evaluate the system performance and other related parameters, such as pressure, temperature, concentration of solution, enthalpy, mass flow rate,

heat flow rate at various state points. The assumptions for the analysis are the same as those for the single-stage-absorption-system given in Sec.(2.2). The details of this analysis are as following :

Condenser pressure ' P_c ', evaporator pressure ' P_e ', generator-II pressure ' P_{g2} ', absorber pressure ' P_a ' and the concentration of LiBr-H₂O solution, leaving the absorber ' X_6 ' are the parameters which can be determined by using the same procedures as were adopted for the single-stage-vapour-absorption-system, described in section (2.2).

In generator-II, the pressure ' P_{g2} ' is now known. Also, the source-temperature for heating the solution in generator-I and the solution temperature in this generator ' T_{g1} ' are known. Hence, in both the components viz generator-I and generator-II, one property only is known for the solution in each component. The thermodynamic principles, however, require the prior knowledge of two independent properties, in order to define the state of solution uniquely. But the determination of second property for the solution in both generator-I and generator-II, presents a problem, which is solved by adopting the following procedure :

First of all, the concentration of LiBr-H₂O solution ' X_{12} ' leaving the generator-II is assumed. Since, the ' P_{g2} '

in the generator-II is already known, ' P_{g2} ' and ' X_{g2} ' i.e. ' X_{12} ' are the two properties which fix the thermodynamic state of the solution. Therefore, the solution temperature leaving generator-II can be determined by using Appendix-2. Assuming a terminal temperature difference of 5°C between the refrigerant vapour coming from generator-I ' T_{r1} ' and the solution temperature ' T_{g2} ', for heat-transfer between the two fluids, ' T_{r1} ' can also be determined. Using Appendix-2, the vapour pressure of the refrigerant corresponding to temperature ' T_{r1} ' can be directly read. This pressure ' P_{g1} ' corresponds to the pressure in generator-I. Since ' T_{g1} ' is already known, the concentration of the LiBr- H_2O solution ' X_{10} ', leaving generator-I can be evaluated with the help of Appendix-2.

We now use the mass, material (LiBr), heat and energy balances in the following components of the system. The details of which are presented below :

(a) Generator-I

Mass balance :

$$\dot{m}_8 = \dot{m}_1 + \dot{m}_{10} \quad (3.1)$$

Material balance (LiBr Conservation) :

$$\dot{m}_8 X_8 = \dot{m}_{10} X_{10} \quad (3.2)$$

(b) Generator-II

Mass balance :

$$\dot{m}_{11} = \dot{m}_{12} + \dot{m}_9 \quad (3.3)$$

Material balance :

$$\dot{m}_{11} X_{11} = \dot{m}_{12} X_{12} \quad (3.4)$$

Energy balance :

$$\dot{m}_1 (H_1 - H_2) = \dot{m}_{12} H_{12} + \dot{m}_9 H_9 - \dot{m}_{11} H_{11} \quad (3.5)$$

(c) Heat-Exchanger-I

Heat balance :

$$\dot{m}_6 (H_7 - H_6) = \dot{m}_{12} (H_{12} - H_{13}) \quad (3.6)$$

Heat exchanger effectiveness :

$$\begin{aligned} \eta_1 &= \frac{\text{Useful enthalpy drop}}{\text{Maximum possible enthalpy drop}} \\ &= \frac{H_{12} - H_{13}}{H_{12} - H_6} \end{aligned} \quad (3.7)$$

(d) Heat-Exchanger-II

Heat balance :

$$m_7 (H_8 - H_7) = m_{10} (H_{10} - H_{11}) \quad (3.8)$$

Heat exchanger effectiveness

$$\eta_2 = \frac{H_{10} - H_{11}}{H_{10} - H_7} \quad (3.9)$$

(e) Absorber

Mass balance :

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_{13} \quad (3.10)$$

Material balance :

$$\dot{m}_6 X_6 = \dot{m}_{13} X_{13} \quad (3.11)$$

(f) Condenser

Mass balance :

$$\dot{m}_9 + \dot{m}_2 = \dot{m}_3 \quad (3.12)$$

In expressions (3.1) to (3.12), the parameters

' H_1 ', ' H_2 ', ' H_5 ', ' H_6 ', ' H_9 ', ' H_{10} ', ' H_{12} ', ' X_6 ', ' n_1 ' and ' n_2 ' are known.

When these known parameters are substituted in the above expressions, they generate a set of six independent equations in six unknown variables, which are as following :

$$\dot{m}_1 (H_1 - H_2) = \dot{m}_{12} H_{12} + \dot{m}_9 H_9 - m_{11} H_{11} \quad (3.13)$$

$$\dot{m}_7 (H_8 - H_7) = \dot{m}_{10} (H_{10} - H_{11}) \quad (3.14)$$

$$\dot{m}_{12} (H_{12} - H_{13}) = \dot{m}_7 (H_7 - H_6) \quad (3.15)$$

$$\dot{m}_{11} = \dot{m}_9 + \dot{m}_{12} \quad (3.16)$$

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_9 \quad (3.17)$$

$$H_{10} - H_{11} = \eta_2 (H_{10} - H_7) \quad (3.18)$$

The unknown variables in the expressions (3.13) to (3.18) are :

' \dot{m}_1 ', ' \dot{m}_9 ', ' \dot{m}_{12} ', ' H_7 ', ' H_8 ' and ' H_{11} '

Simultaneous solution of equations (3.13) to (3.18) is then obtained with the help of MATRIX INVERSION METHOD . This solution yields all the unknown parameters.

Using expressions (3.1) and (3.2), the Generator-Recirculation-Ratio, 'GRR' defined in Section (2.2), for the present system is given by ,

$$\begin{aligned} \text{GRR} &= \frac{\text{Mass of LiBr-H}_2\text{O solution, leaving the generator-I}}{\text{Mass of water-vapour driven off from the generator-I}} \\ &= \frac{\dot{m}_{10}}{\dot{m}_1} \\ X &= \frac{x_8}{x_{10} - x_8} \\ \therefore \frac{\dot{m}_{10}}{\dot{m}_1} &= \frac{x_8}{x_{10} - x_8} \quad (3.19) \end{aligned}$$

All the variables in the above expression are known except \dot{m}_{10} , which can be easily determined.

Expression (3.4) for LiBr balance in generator-II is represented as :

$$\dot{m}_{11} X_{11} = \dot{m}_{12} X_{12} \quad (3.20)$$

Since,

$$\dot{m}_{11} = \dot{m}_{10} ,$$

and ,

$X_{11} = X_{10}$, the expression (3.20) reduces to ,

$$\dot{m}_{10} X_{10} = \dot{m}_{12} X_{12}$$

or,

The concentration X_{12} of the LiBr-H₂O solution is given by,

$$X_{12} = \frac{\dot{m}_{10} X_{10}}{\dot{m}_{12}} \quad (3.21)$$

All the parameters on the R.H.S. of the above expression are known and, hence, using (3.21), X_{12} can be determined.

This calculated value of ' X_{12} ' is then compared with the value of this parameter, assumed to be known initially. If they are not in close agreement with each other, then a different value of ' X_{12} ' is assumed and the above procedure is repeated. till the difference between the assumed and calculate value of ' X_{12} ' becomes negligible.

The system performance, represented by C.O.P. is then obtained as :

$$\begin{aligned} \text{C.O.P.} &= \frac{\text{Refrigerating effect in the evaporator}}{\text{Heat supplied in the generator-I}} \\ &= \frac{\dot{m}_4 (H_5 - H_4)}{\dot{m}_{10} H_{10} + \dot{m}_1 H_1 - \dot{m}_8 H_8} \end{aligned} \quad (3.22)$$

The overall heat-balance of the complete system is performed, which corroborates the correctness of the solution-method presented above. The heat balance equations for various components are :

(i) Heat addition in generator-I :

$$q_{g1} = \dot{m}_{10} H_{10} + \dot{m}_1 H_1 - \dot{m}_8 H_8 \quad (3.23)$$

(ii) Heat transfer in generator-II :

$$\begin{aligned} q_{g2} &= \dot{m}_1 (H_1 - H_2) \\ &= \dot{m}_{12} H_{12} + \dot{m}_9 H_9 - \dot{m}_{11} H_{11} \end{aligned} \quad (3.24)$$

(iii) Heat rejection in condenser :

$$q_c = \dot{m}_9 H_9 + \dot{m}_2 H_2 - \dot{m}_3 H_3 \quad (3.25)$$

(iv) Heat extracted in the evaporator:

$$q_e = \dot{m}_4 (H_5 - H_4) \quad (3.26)$$

(v) Heat rejection in the absorber :

$$q_a = \dot{m}_{13} H_{13} + \dot{m}_5 H_5 - \dot{m}_6 H_6 \quad (3.27)$$

The final heat balance is given by,

Total heat inflow into the system

= Total heat outflow from the system

or,

$$q_e + q_{g1} = q_c + q_a \quad (3.28)$$

This completes the thermodynamic analysis of the proposed double-stage-vapour-absorption-system.

3.3 COMPUTER MODELLING :

A computer programme package given in Appendix-5 has been developed on the basis of the thermodynamic analysis given above. The complete logic, which forms the basis for the development of the programme has been mentioned in the flow-chart, given in Fig. (3.1). Various subroutines developed to solve state equations of LiBr-H₂O solution and refrigerant along with other parameters are given in Section (3.3.1).

Newton-Raphson's Numerical Technique (Sec. 3.3.2) has been used to solve the non-linear algebraic equations in the problem. Subroutines, given by Numerical Algorithm Group (NAG), are employed to solve the simultaneous linear equations. The programme has been made very general to encompass a class of similar problems, which can be solved by simply changing input variables in the parent programme.

3.3.1 Subroutines for solving state equations :

The various subroutines developed for solving the thermodynamic state equations (Appendix-1) for the LiBr-H₂O solution and refrigerant are given below :

- (i) subroutine for the pressure of saturated refrigerant vapour when its temperature is known :

$$P = P (TR) \quad (3.29)$$

- (ii) subroutine for the concentration of the solution when its temperature and refrigerant temperatures are known :

$$X = X (TS, TR) \quad (3.30)$$

- (iii) subroutines for the concentration of the solution when refrigerant temperature and pressure are known :

$$X = X (P, TR) \quad (3.31)$$

- (iv) subroutine for enthalpy of solution, when solution temperature and concentration are known.

$$H = H (TS, X) \quad (3.32)$$

- (v) subroutine for the solution temperature when its enthalpy and concentration are known.

$$TS = TS (H, X) \quad (3.33)$$

3.3.2 Algorithm, Newton-Raphson Technique :

The Newton-Raphson iterative technique is used for solving higher-order, non-linear algebraic equations, as follows :

A function $f(X)$, which is continuous and has a continuous derivative is given. Also given is a starting value X_0 (the concentration of LiBr-H₂O solution in the present case).

For $n = 1, 2, \dots$ until termination, do :

compute $f'(X_n)$

If $f'(X_n) = 0$, signal and stop.

Else, compute

$$X_{n+1} = X_n - \frac{f(X_n)}{f'(X_n)} \quad (3.34)$$

till difference between the values X_{n+1} and X_n tends to zero.

The termination criterion used for iteration :

After N steps (N assumed, fixed)

or,

$$|X_{n+1} - X_n| \leq \epsilon \quad (\epsilon > 0, \text{ given})$$

or,

$$|f(X_n)| \leq \alpha \quad (\alpha > 0 \text{ given})$$

Stop.

The flow-chart for the analysis is given in Fig. (3.1).

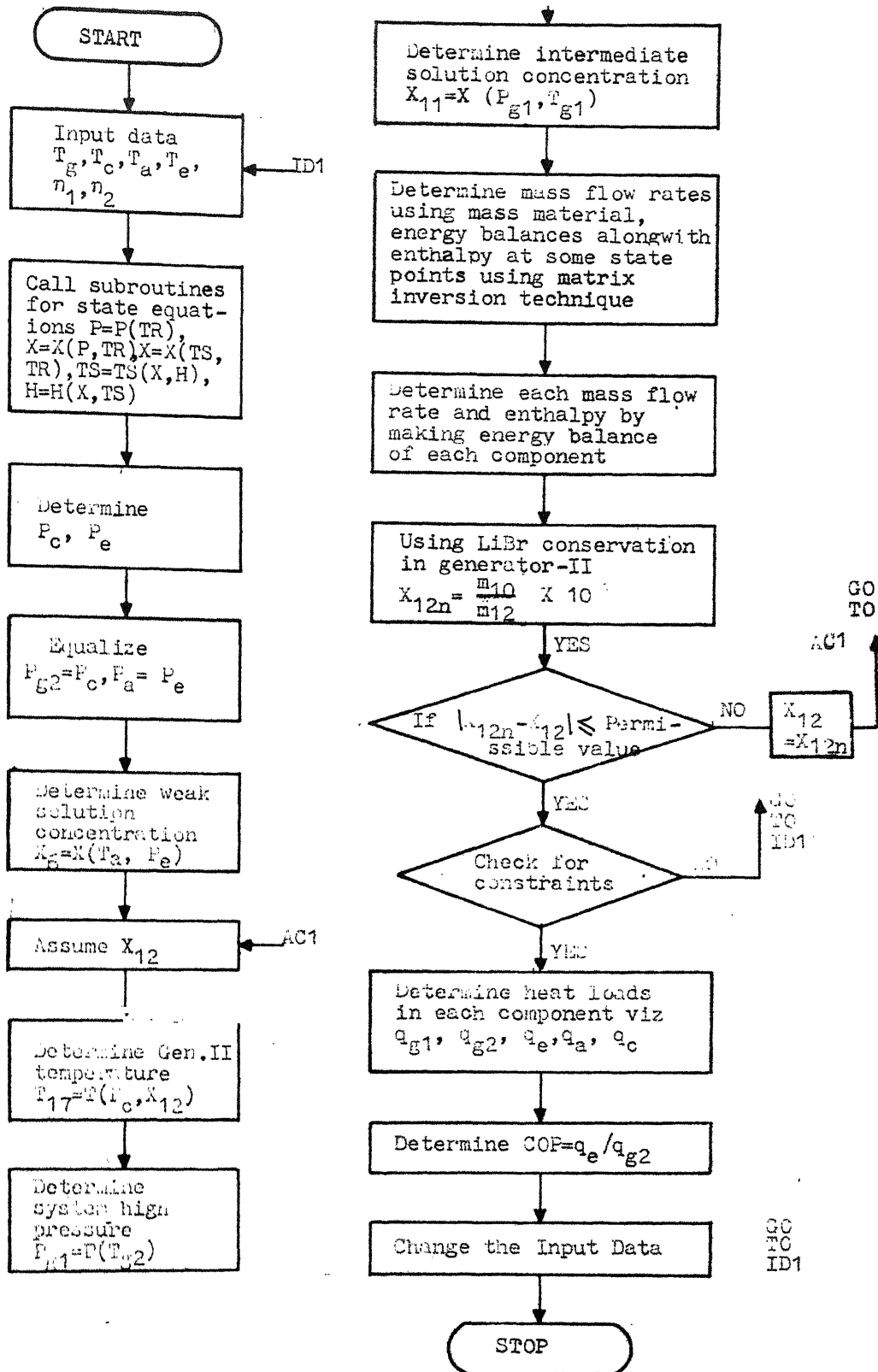


FIG.3.1 FLOW-CHART FOR COMPUTER MODELLING OF A DOUBLE-STAGE-VAPOUR-ABSORPTION-SYSTEM

CHAPTER- IV

ECONOMICS OF VAPOUR ABSORPTION AND VAPOUR-COMPRESSION SYSTEMS

OPERATIONAL COST :

A comparative cost-analysis is carried out to study the relative economics of the present system with its most competing alternative i.e. the-vapour-compression-system. As a case-study, the air-conditioning unit, installed at I.I.T. Kanpur which operates on a vapour-compression -cycle having R-22 as refrigerating fluid has been considered. The detailed data for the air-conditioning unit are presented in Appendix 6. The total cost of any of the above two system involves the following two costs :

- (i) first cost,
- (ii) running or operational cost.

It has been reported.in Reference [20_7] that the first cost of the two system is almost the same. We shall, therefore, compare only the running cost of the two systems.

The running cost of the vapour-compression-system consists of the following :

- (a) Cost of the electrical energy used for pumping cooling-water in the condenser.

Since, the condenser heat load in both the systems vary, the electrical energy used will also be different for the two systems. This has been discussed in detail later.

- (b) Cost of the electrical energy, required for circulating the chilled water in the evaporator.

As the cooling capacity of the evaporator in both the system is taken as the same, the electrical energy used for the purpose, will be the same for both the systems. Hence its contribution has not been considered.

- (c) Cost of the electrical-energy consumed in the compressor

Contribution of this cost is maximum in running the compression-system. Its inclusion for evaluating the total running cost of the system is, therefore a must.

Details of the method for obtaining the total running cost of the vapour-compression-system is presented as follows :

Capacities of the condenser pumps and compressors, used in the system and considered for the case study are reported in (Appendix-6).

Suppose,

PWR_P = Total power of the condenser-pumps [KW]

$$PWR_c = \text{Total power of the compressors } [KW]$$

Hence,

Total power requirement of the complete system will be given by,

$$\text{Total power} = PWR_T = PWR_P + PWR_c \quad (4.1)$$

Annual electrical energy consumption, considering continuous running of the system is then represented as ,

$$(AEC)_{VCS} = PWR_T \times 24 \times 365 \quad [KW-hr] \quad (4.2)$$

In the above expression, 'AEC' represents the 'ANNUAL ENERGY CONSUMPTION' and the subscript 'VCS' stands for the vapour-compression-system.

Let the cost of electrical energy be denoted by 'Cen' in $[Rs/KW.hr]$.

The total cost involved in meeting the running cost of the system annually is, then, given by ,

$$(TRC)_{VCS} = (AEC)_{VCS} \times Cen \quad (4.3)$$

where,

TRC represents the total running cost involved.

The total running-cost of the double-stage-vapour-absorption-system (DSVAS) is then obtained as following :

Factors, affecting the running cost of the system are :

- (1) Cost of the electrical energy, consumed for pumping cooling-water in the condenser.
- (2) Cost of the electrical energy, required for the solution-pump to raise the pressure of the LiBr-H₂O solution from the absorber pressure to generator pressure.

The cost-element, given in (1) is determined as follows :

The mass flow rate of the cooling-water is determined by using the relation :

$$q_c = \dot{m}_w \times C_{pw} \Delta T \quad (4.4)$$

In the above expression, ' q_c ', ' C_{pw} ' and ' ΔT ' are known parameters. ' q_c ' is condenser heat load obtained numerically using the computer programme given in Appendix -5; C_{pw} the specific heat of water, is taken from the table given in [18]; ΔT is the allowable temperature rise in the condenser which has been assumed as 6°C in our case (Appendix-6). Hence the unknown parameter \dot{m}_w is determined from the expression (4.4).

The volume flow rate of cooling-water i.e. Q through the condenser is then given by,

$$Q = \frac{\dot{m}_w}{\rho_w} \quad (4.5)$$

The pressure-drop ' ΔP ' through the condenser tube is calculated as following :

$$p = \frac{fL V_w^2}{2 d_i} \rho_w \quad (4.6)$$

In the above expression, ' L ', ' V_w ' and ' f ' represent the length of tubes, velocity of water and the friction factor respectively. Details of the methods for their determination are discussed below:

Length of condenser tubes :

The condenser heat load q_c is given by the relation,

$$q_c = U_o \times A_o \times \text{LMTD}$$

$$\therefore A_o = \frac{q_c}{U_o \times \text{LMTD}} \quad (4.7)$$

$$\text{LMTD} = \frac{\Delta T_i - \Delta T_o}{\ln \frac{\Delta T_i}{\Delta T_o}} \quad (4.8)$$

The overall heat transfer coefficient U_o is given by

$$U_o = \frac{1}{R_o + R_t + R_{ff} + R_i}$$

$$= \frac{1}{\frac{1}{h_{con}} + \frac{b}{k_m} \frac{d_o}{d_m} + \frac{d_o}{h_{ff} d_i} + \frac{1}{h_w} \frac{d_o}{d_i}} \quad (4.9)$$

Where,

h_{con} = the mean condensing coefficient for vapour
condensing on the outside of horizontal tube is

$$= 0.725 \left(\frac{g \rho_w^2 H_{fg} K_w^3}{\mu t N d_o} \right)^{0.25} \quad [9_7] \quad (4.10)$$

K_w = conductivity of water

μ = viscosity of water

N = number of tubes in one vertical row, assumed
to be 3 as explained in [9_7]

d_o = outer dia of tube, assumed to be 18 mm (Appendix-6)

H_{fg} = Latent heat of vaporisation of water

t = temperature difference between the vapour and
surface of the tube (initially assumed for
iteration purpose as 5°C)

h_{ff} = fouling factor taken

$$= 0.000176 \text{ (m}^2\text{K/W) , [Reference-19_7]}$$

d_i = inner dia of tube, assumed to be 16 mm (Appendix-6)

d_m = mean diameter

$$= \frac{d_o + d_i}{2}$$

b = thickness of tube

$$= \frac{d_o - d_i}{2}$$

K_m = conductivity of tube material

h_w = water-side heat transfer coefficient *

After substituting all the parameters in expression (4.9), the overall heat transfer coefficient U_o is obtained.

Now all the terms on the R.H.S. of expression (4.7) are known. Hence A_o can be determined from the above expression.

Also, the condenser load is given by ,

$$q_c = h_{con} \times A_o \times \Delta t \quad (4.11)$$

$$\therefore \Delta t = \frac{q_c}{h_{con} \times A_o} \quad (4.12)$$

If the calculated value of t does not match with its assumed value in expression (4.10), new value of Δt is assumed and the entire procedure to calculate A_o is repeated. The calculations are terminated, when the difference between the assumed and the calculated value of t is negligible.

* The expression for the heat transfer coefficient for fluids flowing inside tubes, i.e. water in this case, is of the form

$$Nu = C (Re)^n (Pr)^m$$

where, n and m are exponents. The constant C and exponents in the above expression are given by,

$$\frac{h_{wdi}}{K_w} = 0.023 \left(\frac{V d_i \rho_w}{\mu} \right)^{0.8} \times \left(\frac{C_p}{K} \right)^{0.8} \quad \text{[Reference-9_7]}$$

The total outer area of tubes, is also given by,

$$A_o = \pi \times d_o \times L \quad (4.13)$$

where,

L is the total length of tubes.

$$\therefore L = \frac{A_o}{\pi d_o} \quad (4.14)$$

Thus, the required total length of the tubes, used in the condenser, is known.

Velocity of water in tube :

The velocity of water V_w is obtained from the relation

$$V_w = \frac{Q}{\frac{\pi}{4} \times d_i^2} \quad (4.15)$$

where, Q is the volume flow rate of water, obtained in expression (4.5).

Friction-factor :

The friction-factor is given by $\sqrt{17.7}$,

$$f = \frac{1}{1.14 + 2 \log \frac{d_i}{\epsilon} - 2 \log \left[1 + \frac{9.3}{\text{Re}(\epsilon/d_i) \sqrt{f}} \right]}^2 \quad (4.16)$$

where,

' ϵ ' is absolute roughness of pipe material given in appendix -7.

Re is given by

$$Re = \frac{V_w d_i \rho_w}{\mu} \quad (4.17)$$

Since, the equation (4.16) is implicit in 'f', Newton-Raphson numerical technique (Sec. 3.4) is adopted to solve this equation and to obtain the value of 'f'.

The pressure drop ' Δp ' can then be easily obtained, using expression (4.6).

The power required for pumping is then given by,

$$(PWR)_{cond} = \Delta p \times Q \quad (4.18)$$

In the above expression, the subscript cond stands for condenser .

The annual cost of the electrical energy represented as $(TRC)_{cond}$ is then expressed as :

$$(TRC)_{cond} = (PWR)_{cond} \times 24 \times 365 \times C_{en} \quad (4.20)$$

where,

' C_{en} ' is the cost of electrical energy in Rs./KW-hr.

The second running cost-element i.e. the cost of electrical energy required for the solution pump to raise the pressure. Referring to Fig. (1.4), the pressure against which

the the solution pump has to work, comprises of the following pressure drops :

- (i) pressure difference between the generator and absorber (Δp_1),
- (ii) pressure drop in solution heat-exchanger-I (Δp_2) and
- (iii) pressure drop in solution heat-exchanger-II (Δp_3)
- (iv) pressure drop in pipe length (Δp_4)

P_1 is determined by using the relation

$$P_1 = P_{g1} - P_a \quad (4.21)$$

where,

P_{g1} = generator-I pressure,

P_a = absorber pressure

P_{g1} and P_a are known from Appendix -5

Δp_2 and Δp_3 are evaluated by following the same procedure as adopted for obtaining Δp of expression (4.1) for condenser cooling-water. The properties of LiBr-H₂O solution, given in Reference [26], are however used now, instead of the properties of water.

The pressure drop Δp_4 in pipe length can be neglected as it is negligible as compared to above pressure drops.

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The total pressure drop which is to be supplied by the solution pump is therefore, given as :

$$\Delta P_T = \Delta P_1 + \Delta P_2 + \Delta P_3 \quad (4.22)$$

From the knowledge of P_T , the power required for solution pump $(PWR)_{SP}$ can be determined as :

$$PWR_{SP} = \Delta P_T \times Q_S \quad (4.23)$$

where,

Q_S = solution flow rate, obtained from Appendix-5.

Total running cost for solution pump can be expressed as :

$$(TRC)_{SP} = (PWR)_{SP} \times 365 \times 24 \times Cen \quad (4.24)$$

The total running cost, incurred annually for the complete system operation of double-stage-vapour-absorption-system is given by,

$$(TRC)_{DSVAS} = (TRC)_{cond} + (TRC)_{SP} \quad (4.25)$$

where,

$(TRC)_{DSVAS}$ stands for the total running cost of the double-stage-vapour-absorption-system, incurred annually.

The total pressure drop which is to be supplied by the solution pump is therefore, given as :

$$\Delta P_T = \Delta P_1 + \Delta P_2 + \Delta P_3 \quad (4.22)$$

From the knowledge of P_T , the power required for solution pump $(PWR)_{SP}$ can be determined as :

$$PWR_{SP} = \Delta P_T \times Q_S \quad (4.23)$$

where,

Q_S = solution flow rate, obtained from Appendix-5.

Total running cost for solution pump can be expressed as :

$$(TRC)_{SP} = (PWR)_{SP} \times 365 \times 24 \times \text{Cen} \quad (4.24)$$

The total running cost, incurred annually for the complete system operation of double-stage-vapour-absorption-system is given by,

$$(TRC)_{DSVAS} = (TRC)_{\text{cond}} + (TRC)_{SP} \quad (4.25)$$

where,

$(TRC)_{DSVAS}$ stands for the total running cost of the double-stage-vapour-absorption-system, incurred annually.

A complete computer-programme on DEC-1090, given in appendix-8, has been developed to solve the above problem and obtain the total running-cost of the two systems. From the programme, we obtain

$$(a) \quad (TRC)_{DSVAS} = \text{Rs. } 3,73,350.00$$

$$(b) \quad (TRC)_{VCS} = \text{Rs. } 8,08,000.00$$

Therefore the annual saving in running cost with the double-stage-vapour-absorption-system as compared to vapour-compression-system is given by,

$$(c) \quad \begin{aligned} \text{Annual saving} &= (TRC)_{VCS} - (TRC)_{DSVAS} \\ &= \text{Rs. } 4,34,650.00 \end{aligned}$$

or

$$(d) \quad \text{Daily saving} = \text{Rs. } 1,190.00$$

$$(e) \quad \text{Percentage saving} = 53.8 \%$$

CHAPTER-V

RESULTS AND DISCUSSION

Computer modelling of a double-stage-vapour-absorption-system, using thermodynamic equations of state has been developed to determine the feasibility and economy of its operation. DEC-1090 computer with FORTRAN as coding language has been used for executing the programme. A numerical model of one-stage-vapour-absorption-system has also been developed to compare its relative performance with that of a double-stage-system.

The results of the above analyses have been presented in Tables 1.11. On the basis of the data obtained from these tables, figures (5.1 - 5.11) have been drawn.

Figures (5.2-5.4) show the variation of C.O.P. with generator-I temperature for both the systems. Also, the minimum temperatures at which double-stage-absorption-system starts operating for different cooling water temperatures, have been shown. It is observed that this system starts working at a minimum temperature of 70°C , when cooling water temperature is 20°C and at about 90°C for cooling water-temperature of 35°C .

An interesting result is observed from Fig. (5.3). The crystallization of $\text{LiBr-H}_2\text{O}$ solution begins at around 91°C for a single-stage-system. But, this is the range of temperature, at which a double-stage-vapour-absorption-system starts working. This establishes the fact that when the single-stage-vapour-absorption-system stops functioning, the double-stage-system starts operating. Figs.(5.2-5.4) show that the maximum value of C.O.P. obtained in the double-stage-system is almost twice the value of the corresponding single-stage-system (~ 1.7 as compared to ~ 0.85). This is due to the fact that about 80-90 % of the energy input in the generator-I is recovered as the heat of condensation in generator-II, to further drive off the refrigerant vapour, which otherwise would have been wasted in the condenser.

As C.O.P. is the index of effectiveness of the system, it depicts that a double-stage-vapour-absorption-system has almost half the energy consumption per unit cooling achieved, in comparison to a single-stage-system.

The variation of generator-I temperature V/S C.O.P. in a double-stage-system shown in Fig. (5.1) reveals that, upto a certain increase in generator-I temperature, C.O.P. also increases and afterwards it starts decreasing. The optimum value of generator-I - temperature for maximum C.O.P. can, therefore, be obtained. The optimum value of generator-I

temperature is different for different condensing and evaporating temperatures. From Figure (5.1), it is obvious that for 20°C as condenser-temperature and 5°C as evaporator temperature, the optimum value of generator-I-temperature is 79°C and for 25°C as condenser temperature and 5°C as evaporator temperature, the optimum generator-I-temperature for maximum C.O.P. is 88°C . Maximum C.O.P. is 1.785 in this case.

C.O.P. of the double-stage-vapour-absorption-system falls with the increase in the absorber-temperature. Reduction in C.O.P. is less, initially, but after a certain value of the absorber-temperature, the curve becomes quite steep. This is due to the fact that at higher absorber-temperature, absorption capacity of $\text{LiBr-H}_2\text{O}$ solution to absorb the refrigerant vapour coming from the evaporator decreases. The reduction in the absorption capacity requires more solution to flow for absorbing a given amount of refrigerant. Therefore, more heat is needed in generator-I to extract the refrigerant from the solution.

Figures (5.5-5.6) show that with the increase in condenser-temperature, C.O.P. decreases. However, the rate of reduction is less than that of reduction in C.O.P. with absorber-temperature Fig. (5.8). It shows that variation in C.O.P. is much more sensitive to the absorber-temperature than the condenser-temperature.

At high condensing temperatures such as at atmospheric temperature of water or air, the double-stage-system performs efficiently (Fig. 5.6). Hence, the cooling of condenser can be performed at atmospheric temperature of water or air. The need of a cooling tower, which is generally used to reduce the cooling-water temperature, can therefore be eliminated.

The effectiveness of the heat exchangers-I and II have a significant effect on the C.O.P. With the increase in the effectiveness, the C.O.P. increases (Fig. 5.10) and varies between 1.325 to 1.778 for the corresponding effectiveness 0.5 to 1.0 of heat exchanger-I and II.

Figure 5.9 shows that generator-I heat load first decreases rapidly with the increase in generator-I-temperature, but afterwards it increases slightly. The inversion point of this curve corresponds to the maximum C.O.P. of the double-stage-system. This amounts to conclude that at the optimum generator-I-temperature, the heat load required to be supplied in generator-I is minimum.

Since the double-stage-system uses low grade heat energy for its operation as compared to high grade mechanical energy used by vapour-compression-system, the former has the potential advantages over the later, as far as the operational costs of the

systems are concerned. For the same system capacity of 550 tons that has been studied in the present work, as an case study, the yearly saving achieved in the running cost of a double-stage-vapour-absorption-system is approximately 4.35 lakhs over the vapour-compression system.

Numbers at top of each column in the tables (1-4) denote the following parameters :

1. Serial No;
2. Generator-I temperature ($^{\circ}\text{C}$) ;
3. Generator-II temperature ($^{\circ}\text{C}$) ;
4. Generator-I load (KW) ;
5. Evaporator load (KW) ;
6. Generator-II load (KW) ;
7. Weak-solution concentration
(Kg LiBr/Kg of solution);
8. Intermediate-solution concentration
(Kg LiBr/Kg of solution);
9. Strong solution concentration
(Kg LiBr/Kg of solution);
10. C.O.P., Double-stage ;
11. C.O.P., Single-stage ;
12. C.O.P., Ideal absorption cycle
- Cry. Crystallization
- ONP Operation not possible

Number at top of each column in the Tables (5-11) denote the following parameters :

- 1 Serial No.
- 2 Variable indicated in the table column
- 3 Generator-II temperature ($^{\circ}\text{C}$) ;

- 4 C.O.P. (Double-stage)
- 5 C.O.P. (Single-stage)
- 6 C.O.P. (Ideal-cycle)

For the Table. 12 the numbers at the top of column denote;

- 1 Serial No.
- 2 Heat-Exchanger Effectiveness
- 3 C.O.P. (Double-stage)
- 4 C.O.P. (Single-stage)

Effect of Generator-I Temperature on operating parameters

($T_c = 20^{\circ}C$, $T_a = 20^{\circ}C$, $T_g = 5^{\circ}C$, $\eta_1 = \eta_2 = 0.90$)											
1	2	3	4	5	6	7	8	9	10	11	12
1	52.0									.8395	
2	58.0									.8349	
3	64.0									.8314	
4	70.0	47.1	9.985	17.747	9.064	45.76	48.19	50.60	1.77	.8295	2.702
5	75.0	49.2	9.952	17.747	9.081	45.76	48.75	51.70	1.780	.8276	2.839
6	76.0	51.0	9.942	17.747	9.097	45.76	49.20	52.80	1.785	Cry.	2.974
7	79.0	52.8	9.945	17.747	9.111	45.76	49.76	49.64	53.90	1.786	3.107
8	82.0	55.0	9.946	17.747	9.122	45.76	50.14	54.90	1.785		3.237
9	85.0	57.0	9.954	17.747	9.132	45.76	50.60	55.90	1.781		3.365
10	88.0	59.0	9.966	17.747	9.143	45.76	51.02	56.90	1.779		3.491
11	91.0	60.6	9.975	17.747	9.149	45.76	51.55	57.80	1.778		3.615
12	94.0	62.6	9.994	17.747	9.161	45.76	51.89	57.90	1.776		3.737
13	97.0	64.7	10.008	17.747	9.167	45.76	52.55	59.70	1.772		3.857
14	100.0	66.4	10.025	17.747	9.174	45.76	52.77	60.60	1.770		3.975

($T_e = 25^\circ\text{C}$, $T_a = 20^\circ\text{C}$, $T_e = 5^\circ\text{C}$, $\eta_1 = \eta_2 = 0.90$)

1	2	3	4	5	6	7	8	9	10	11
1	52.0									.8277 1.
2	58.0									.8271 1.
3	64.0				ONP					.8247 1.
4	70.0	48.8	10.378	17.594	9.067	45.76	47.06	48.20	1.695	.8216 1.
5	73.0	50.7	10.204	17.594	9.087	45.76	47.60	49.40	1.724	.8210 1.
6	76.0	52.7	10.109	17.594	9.099	45.76	48.17	50.50	1.740	.8196 1.
7	79.0	54.5	10.069	17.594	9.109	45.76	48.69	51.60	1.747	.8180 2.
8	82.0	56.6	10.054	17.594	9.120	45.76	49.16	52.70	1.750	Cry 2.
9	85.0	58.5	10.041	17.594	9.128	45.76	49.68	53.70	1.752	2.5
10	88.0	60.4	10.040	17.594	9.135	45.76	50.17	54.70	1.752	2.6
11	91.0	62.1	10.047	17.594	9.144	45.76	50.62	55.70	1.751	2.7
12	94.0	64.1	10.059	17.594	9.153	45.76	51.03	56.70	1.749	2.8
13	97.0	66.1	10.074	17.594	9.163	45.76	51.40	57.70	1.746	2.8
14	100.0	68.0	10.086	17.594	9.168	45.76	51.86	58.60	1.742	2.9

TABLE -3

Effect of Generator-I Temperature on operating parameters

$(T_c = 30^{\circ}\text{C}, \quad T_g = 25^{\circ}\text{C}, \quad T_e = 5^{\circ}\text{C}, \quad \eta_1 = \eta_2 = 0.90)$											
1	2	3	4	5	6	7	8	9	10	11	1
1	55.0									.7360	1.1
2	61.0									.7998	1.1
3	67.0									.8086	1.3
4	73.0									.8115	1.45
5	79.0									.8093	1.70
6	82.0	56.3	12.628	17.442	8.882	49.34	49.56	49.30	0.666	.8080	1.78
7	85.0	58.0	12.125	17.442	8.940	49.34	49.85	50.40	1.438	.8052	1.86
8	88.0	59.7	10.901	17.442	9.026	49.34	50.43	51.40	1.600	Cry.	1.941
9	91.0	61.5	10.558	17.442	9.058	49.34	50.96	52.40	1.652		2.016
10	94.0	63.4	10.408	17.442	9.079	49.34	51.45	53.40	1.676		2.091
11	97.0	65.7	10.333	17.442	9.097	49.34	51.90	54.40	1.690		2.164
12	100.0	67.0	10.271	17.442	9.106	49.34	52.42	55.30	1.698		2.236
13	103.0	69.8	10.257	17.442	9.122	49.34	52.80	56.30	1.701		2.307
14	106.0	70.6	10.238	17.442	9.131	49.34	53.23	57.20	1.704		2.412

Effect of Generator-I Temperature on operating parameters

		$(T_c = 35^\circ\text{C},$			$T_a = 30^\circ\text{C},$			$T_e = 2^\circ\text{C},$			$\eta_1 = \eta_2 = 0.90)$	
1	2	3	4	5	6	7	8	9	10	11		
1	64.0										.2897	
2	70.0										.7844	
3	76.0										.7968	
4	82.0										.7986	
5	88.0										.7976	1.52
						ONP						
6	91.0	62.9	17.526	17.328	8.678	50.39	50.20	50.00	0.746	.7967	1.59	
7	94.0	64.1	14.076	17.328	8.762	50.39	50.70	51.00	1.231	Cry.	1.67	
8	97.0	66.6	11.446	17.328	8.964	50.39	51.22	52.00	1.514		1.74	
9	100.0	68.3	10.880	17.328	9.018	50.39	51.70	53.00	1.593		1.81	
10	103.0	70.0	10.589	17.328	9.041	50.39	52.25	53.90	1.636		1.95	
11	106.0	71.9	10.489	17.328	9.064	50.39	52.65	54.90	1.652		2.02	
12	109.0	73.7	10.405	17.328	9.077	50.39	53.13	55.80	1.665		2.08	
13	112.0	75.6	10.359	17.328	9.089	50.39	53.57	56.70	1.675		2.15	

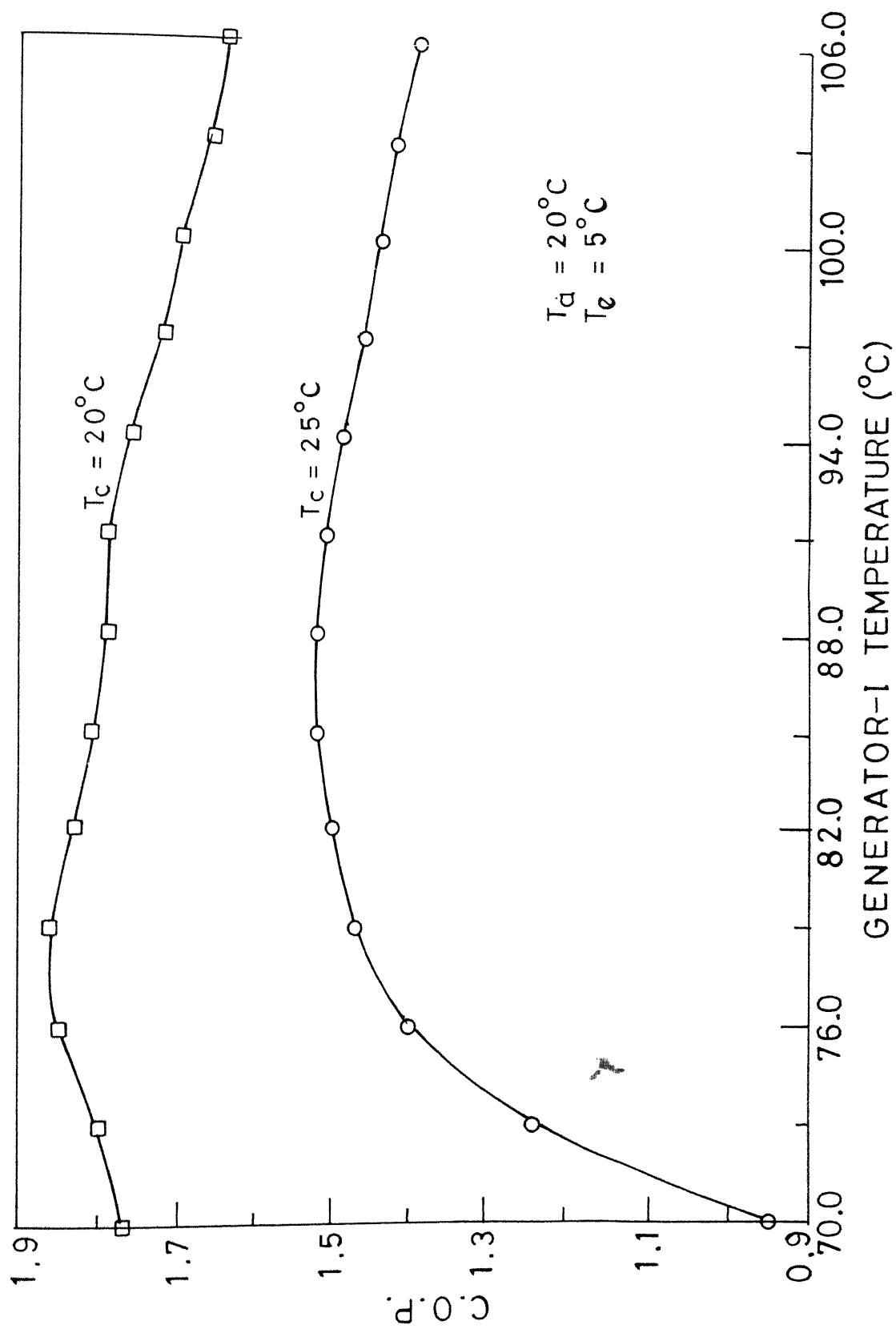


FIG. 5.1 VARIATION OF C.O.P. WITH GENERATOR-I TEMPERATURE.

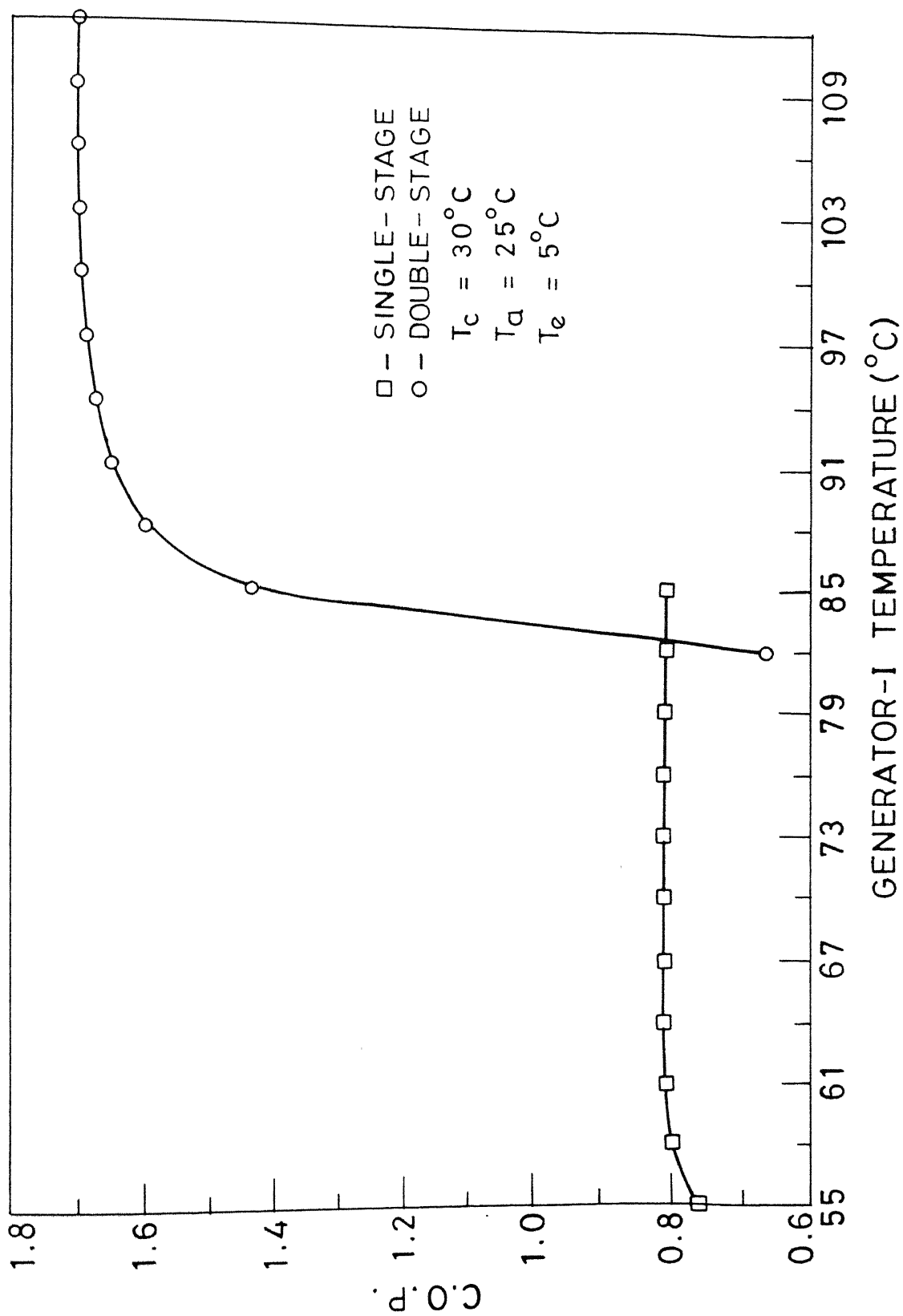


FIG.5.2 VARIATION OF C.O.P. WITH GENERATOR-I TEMPERATURE.

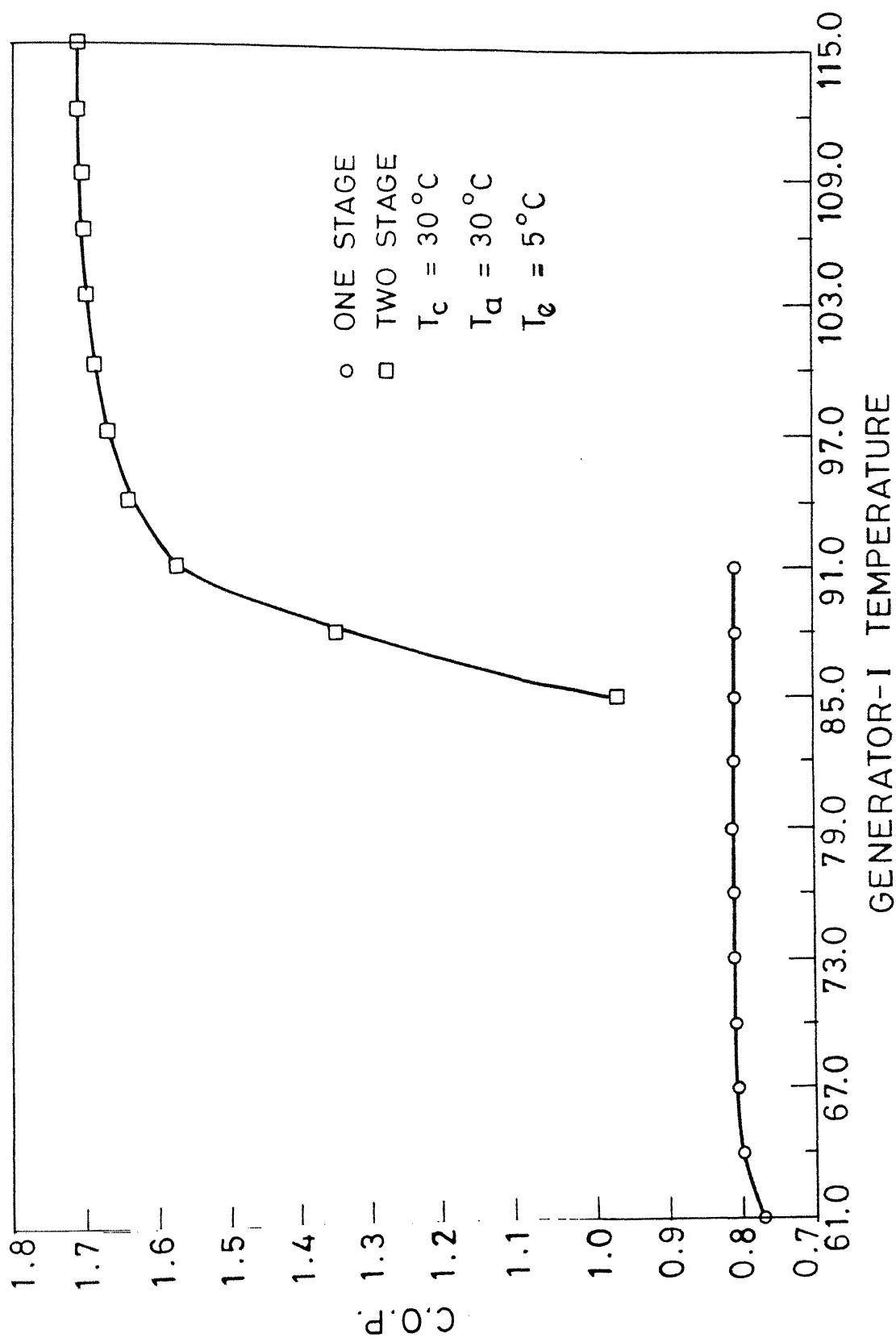


FIG. 5.3 VARIATION OF C.O.P. WITH GENERATOR-I TEMPERATURE.

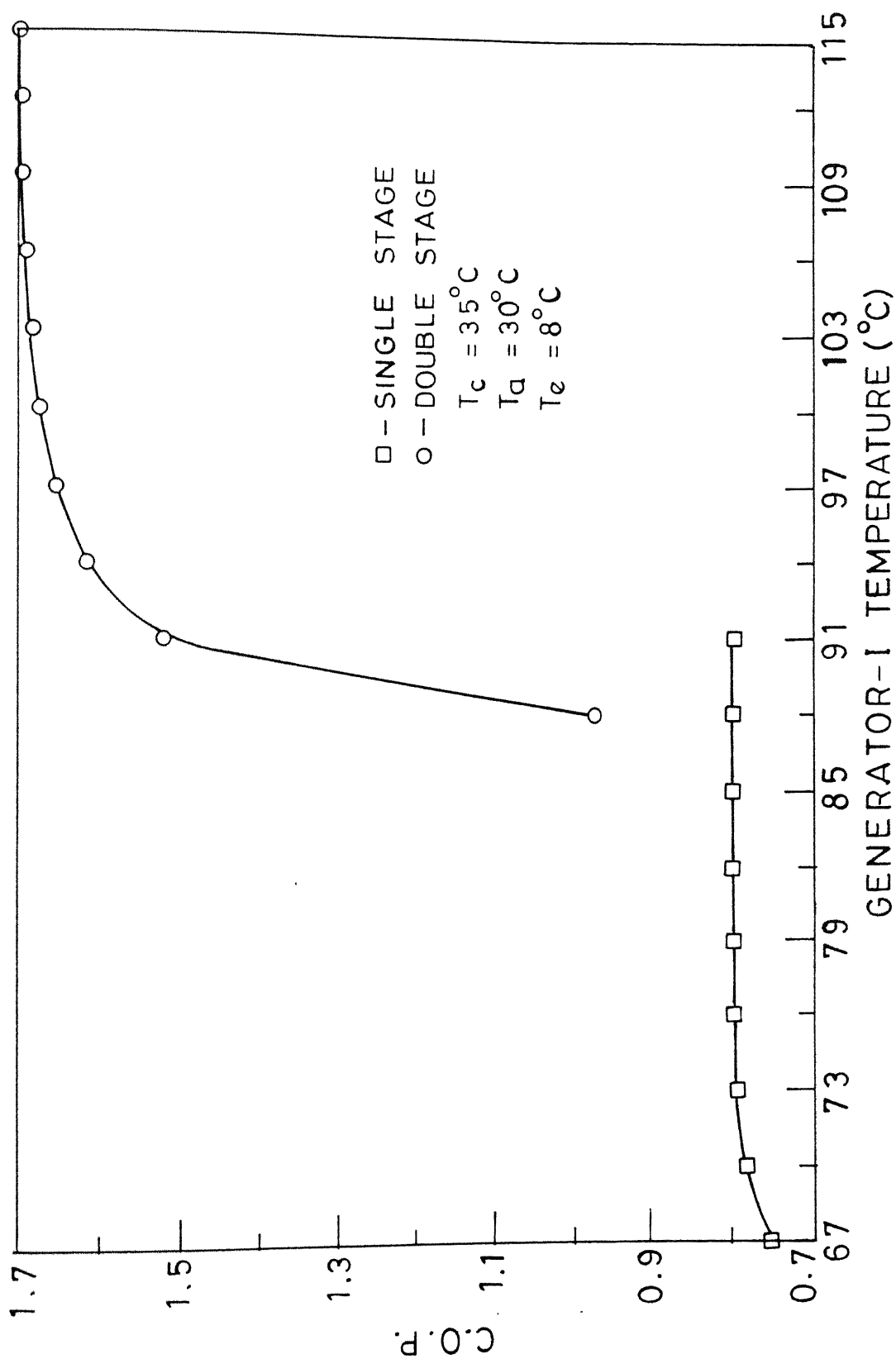


FIG.5.4 VARIATION OF C.O.P. WITH GENERATOR-I TEMPERATURE.

TABLE -5Effect of Condenser-Temperature on C.O.P.

($T_g = 90^{\circ}\text{C}$, $T_a = 25^{\circ}\text{C}$, $T_e = 8^{\circ}\text{C}$, $n_1 = n_2 = 0.90$)						
1	$T_c (^{\circ}\text{C})$	3	4	5	6	7
1	25.0	60.73	1.758	Cry.	2.732	
2	26.0	60.91	1.753	Cry.	2.580	
3	27.0	61.29	1.745	,,	2.445	
4	28.0	61.67	1.736	,,	2.323	
5	29.0	62.05	1.727	,,	2.212	
6	30.0	62.26	1.720	,,	2.111	
7	31.0	62.65	1.709	,,	2.020	
8	32.0	63.05	1.697	Cry.	1.935	
9	33.0	63.28	1.687	.809	1.858	
10.	34.0	63.69	1.671	.807	1.787	
11	35.0	64.05	1.658	.805	1.720	

TABLE -6

Effect of Condenser-Temperature on C.O.P.

($T_g = 95^{\circ}\text{C}$, $T_a = 25^{\circ}\text{C}$, $T_e = 8^{\circ}\text{C}$, $n_1 = n_2 = 0.90$)

1	$T_c (^{\circ}\text{C})$	3	4	5	6	7
1	25.0	63.88	1.759	Cry.	3.145	
2	26.0	64.23	1.753	,,	2.970	
3	27.0	64.58	1.745	,,	2.813	
4	28.0	64.74	1.740	,,	2.673	
5	29.0	65.10	1.733	,,	2.546	
6	30.0	65.46	1.725	,,	2.430	
7	31.0	65.83	1.717	,,	2.324	
8	32.0	66.20	1.708	,,	2.227	
9	33.0	66.39	1.702	,,	2.138	
10	34.0	66.78	1.691	,,	2.056	
11	35.0	67.05	1.681	Cry.	1.980	

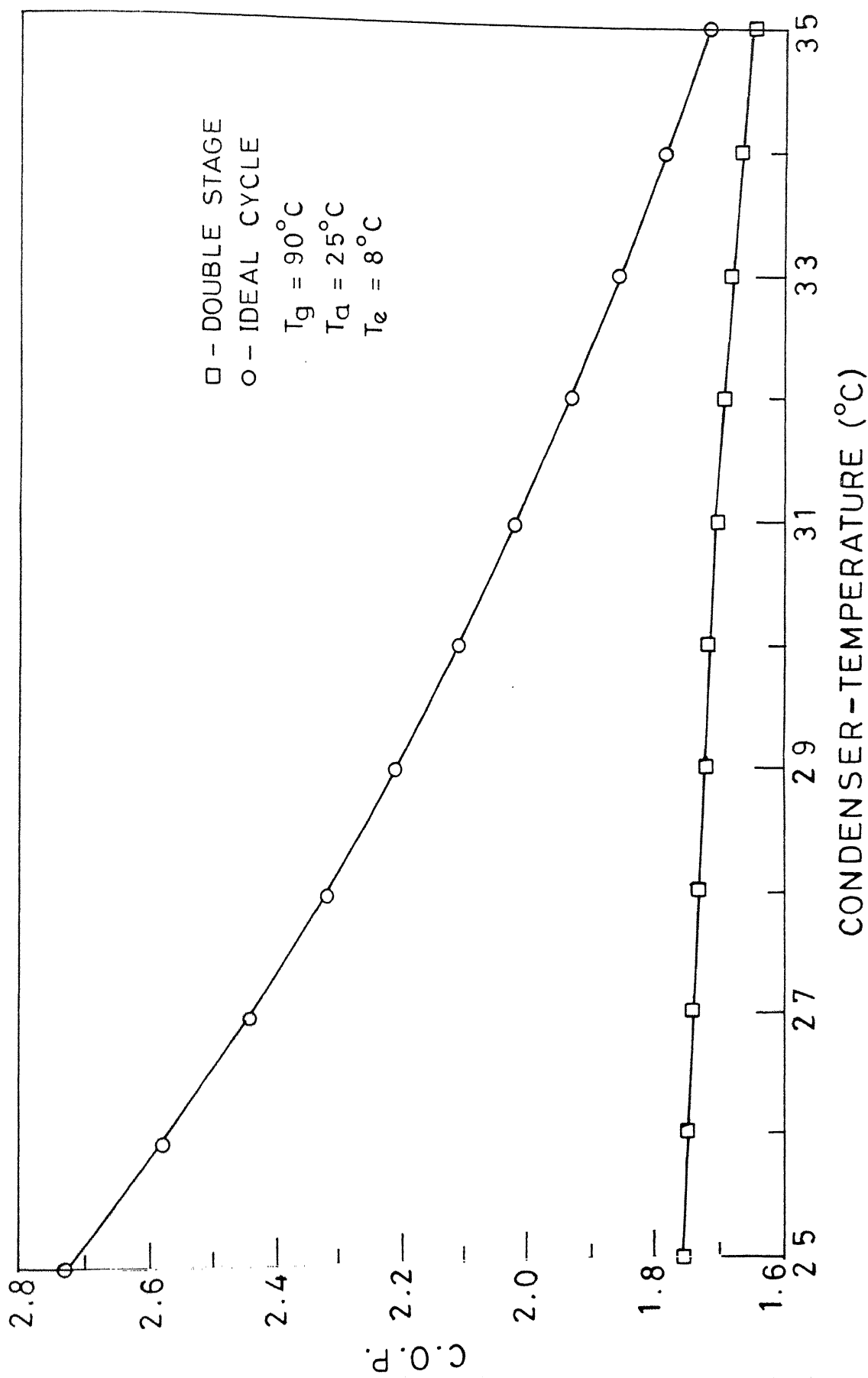


FIG. 5.5 VARIATION OF C.O.P. WITH CONDENSER-TEMPERATURE.

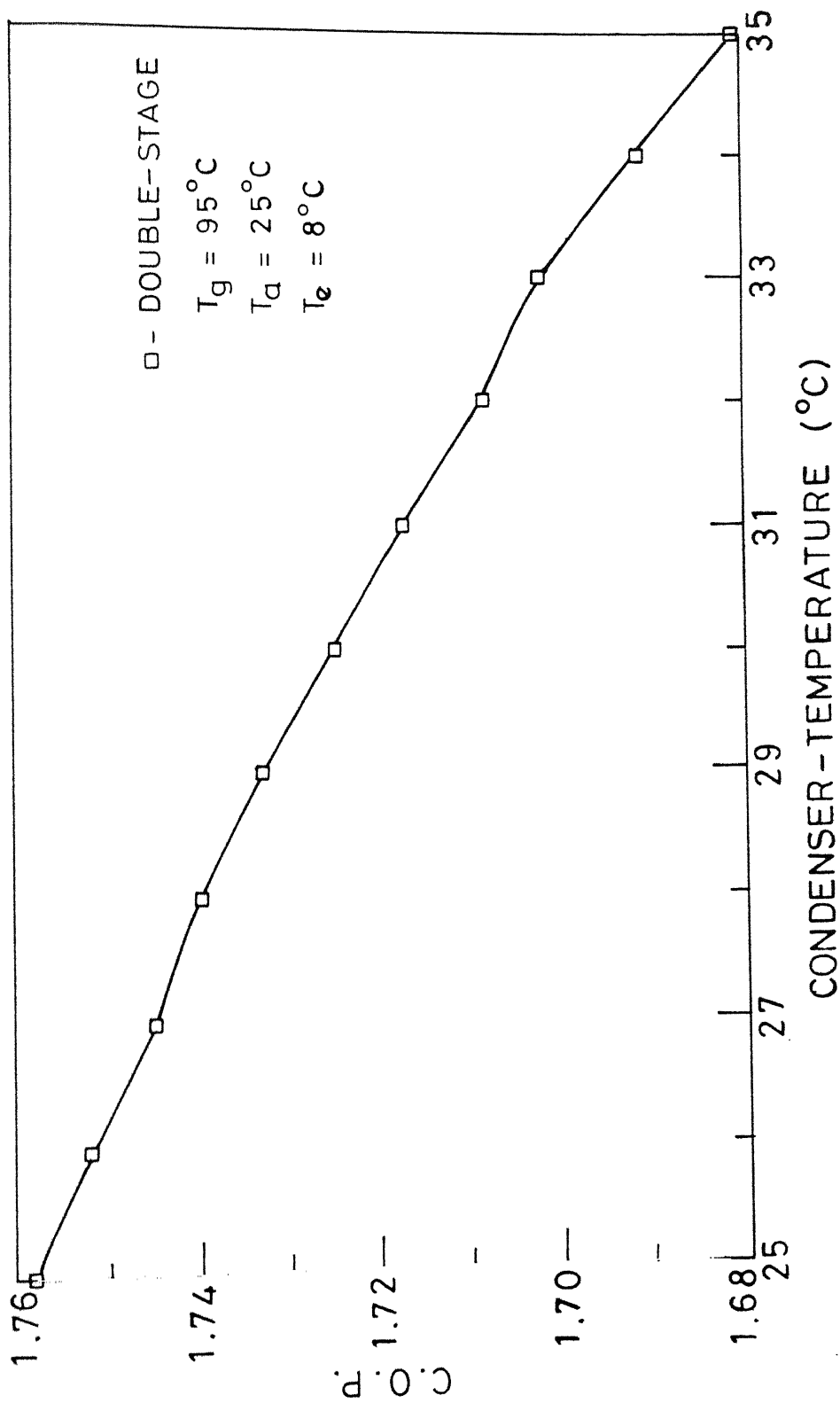


FIG.5.6 VARIATION OF C.O.P. WITH CONDENSER-TEMPERATURE.

TABLE- 7

Effect of Evaporator-Temperature on C.O.F.

(T _g =85°C , T _c = 25°C , T _a =20°C , $\eta_1=\eta_2=0.90$)						
1	T _e (°C)	3.	4	5	6	7
1	3.0	57.34	1.728	Cry.	2.278	
2	4.0	57.71	1.743	,,	2.395	
3	5.0	58.26	1.753	,,	2.524	
4	6.0	58.83	1.760	,,	2.666	
5	7.0	59.58	1.765	,,	2.825	
6	8.0	60.16	1.771	,,	3.001	
7	9.0	60.93	1.775	,,	32.00	
8	10.0	61.23	1.780	,,	3.426	

TABLE- 8

Effect of Evaporator-Temperature on C.O.P.
 $(T_g=95^{\circ}\text{C} , \quad T_c=25^{\circ}\text{C} , \quad T_a=20^{\circ}\text{C} , \quad \eta_1=\eta_2=0.90)$

1	$T_e(^{\circ}\text{C})$	3	4	5	6	7
1	3.0	63.48	1.737	Cry.	2.557	
2	4.0	64.09	1.743	,,	2.688	
3	5.0	64.69	1.749	,,	2.833	
4	6.0	65.30	1.754	,,	2.993	
5	7.0	66.12	1.757	,,	3.171	
6	8.0	66.95	1.760	,,	3.369	
7	9.0	67.78	1.762	,,	3.592	
8	10.0	68.62	1.765	,,	3.846	

TABLE- 9

Effect of Evaporator-Temperature on C.O.P.
 $(T_g=90^{\circ}\text{C} , \quad T_c=30^{\circ}\text{C} , \quad T_a=25^{\circ}\text{C} , \quad \eta_1=\eta_2=0.90)$

1	$T_e(^{\circ}\text{C})$	3	4	5	6	7
1	3.0	60.26	1.339	Cry	1.831	
2	4.0	60.62	1.554	,,	1.908	
3	5.0	60.98	1.635	,,	1.991	
4	6.0	61.34	1.678	,,	2.082	
5	7.0	61.89	1.700	,,	2.180	
6	8.0	62.26	1.720	,,	2.287	
7	9.0	62.81	1.732	,,	2.405	
8	10.0	63.37	1.741	,,	2.534	

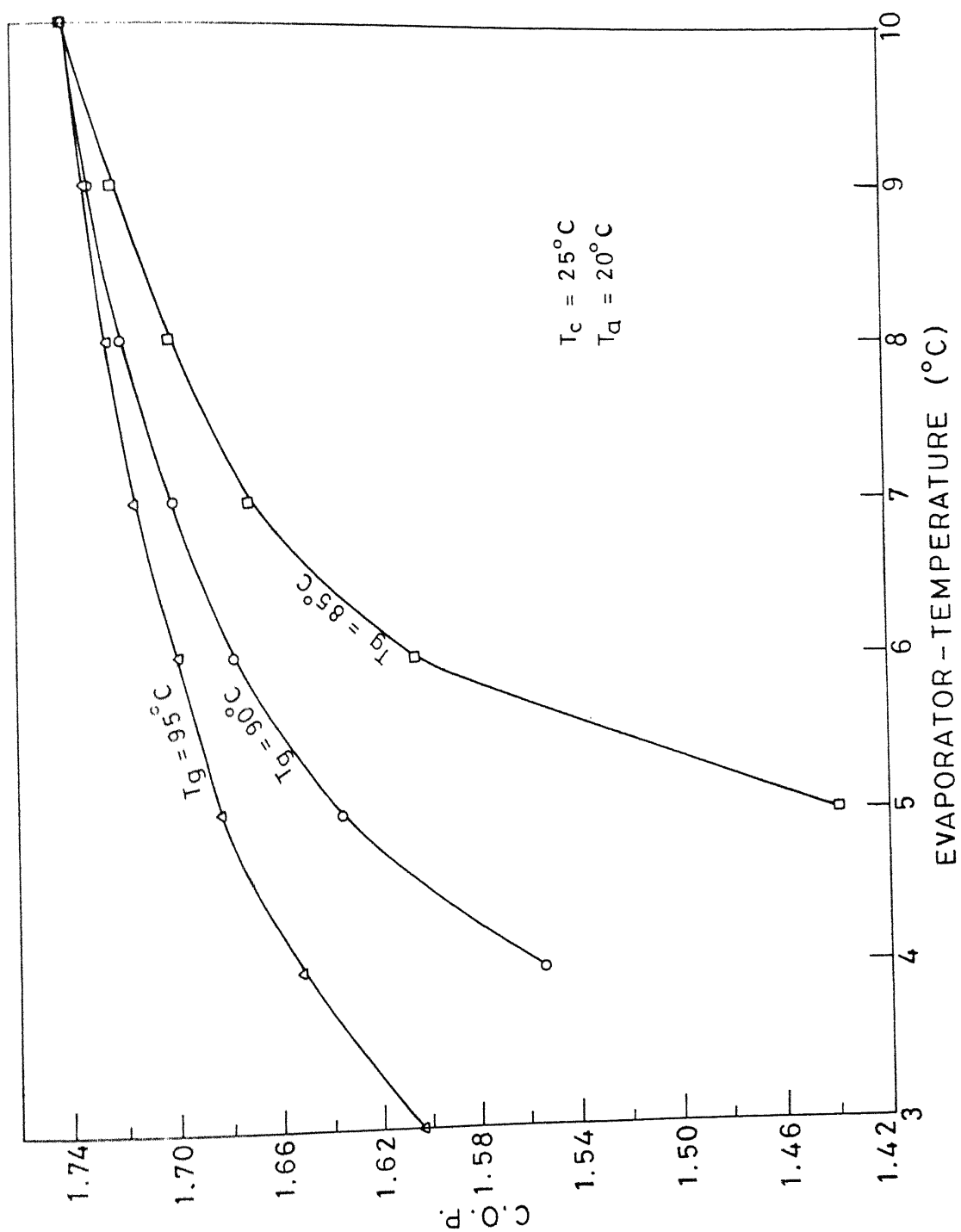


FIG. 5.7 VARIATION OF C.O.P. WITH EVAPORATOR-TEMPERATURE (DOUBLE-STAGE).

TABLE- 10

Effect of Absorber-Temperature on the C.O.P.

($T_c=90^{\circ}\text{C}$, $T_c=30^{\circ}\text{C}$, $T_e=5^{\circ}\text{C}$, $n_1=n_2=0.90$)						
1	$T_a(^{\circ}\text{C})$	3	4	5	6	7
1	20.0	63.05	1.719	Cry	2.145	
2	21.0	62.63	1.711	,,	2.114	
3	22.0	62.07	1.704	,,	2.083	
4	23.0	61.71	1.690	,,	2.052	
5	24.0	61.34	1.669	,,	2.022	
6	25.0	60.98	1.635	,,	1.991	
7	26.0	60.62	1.577	,,	1.960	
8	27.0	60.26	1.452	,,	1.930	
9	28.0	59.91	1.027	,,	1.899	
10	30.0	ONP		.8072	1.869	

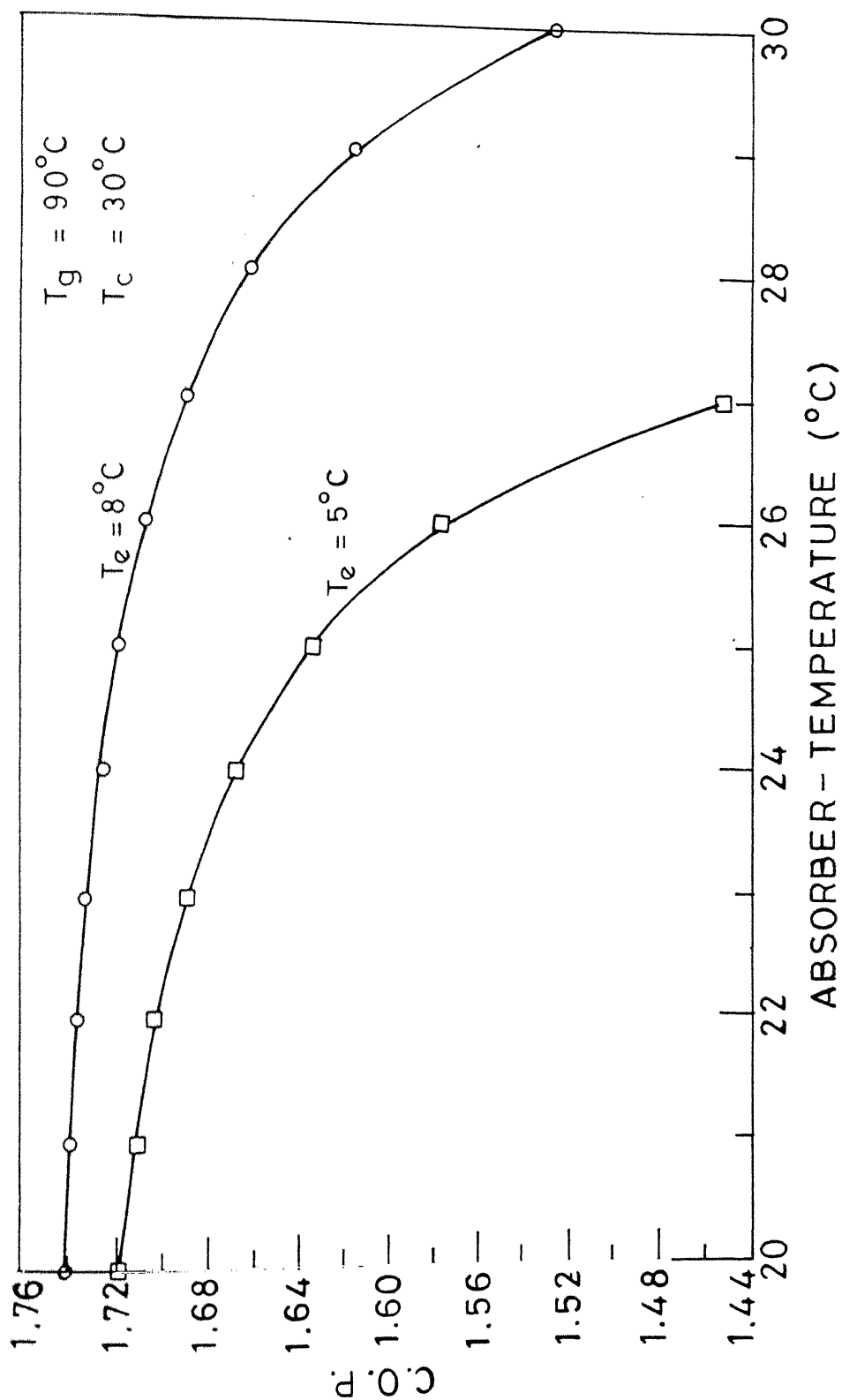


FIG.5.8 VARIATION OF C.O.P. WITH ABSORBER-TEMPERATURE (DOUBLE STAGE).

TABLE- 11

Effect of Absorber-Temperature on the C.O.P.

($T_g=90^{\circ}\text{C}$, $T_c=30^{\circ}\text{C}$, $T_e=8^{\circ}\text{C}$, $\eta_1=\eta_2=0.90$)						
1	$T_a(^{\circ}\text{C})$	3	4	5	6	7
1	20.0	65.24	1.744	Cry	2.518	
2	21.0	64.88	1.741	,,	2.463	
3	22.0	64.31	1.739	,,	2.428	
4	23.0	63.75	1.736	,,	2.393	
5	24.0	63.18	1.733	,,	2.358	
6	25.0	62.88	1.726	,,	2.322	
7	26.0	62.26	1.720	,,	2.287	
8	27.0	61.89	1.708	,,	2.252	
9	28.0	61.52	1.690	,,	2.217	
10	29.0	61.16	1.663	,,	2.182	
11	30.0	60.80	1.524	,,	2.111	

TABLE -12

Effect of Heat-Exchanger effectiveness on C.O.P.

($T_g=90^{\circ}\text{C}$, $T_c=25^{\circ}\text{C}$, $T_a=20^{\circ}\text{C}$, $T_e=5^{\circ}\text{C}$)			
1	η	3	4
1	0.50	61.12	1.325
2	0.55	61.12	1.392
3	0.60	61.315	1.452
4	0.65	61.315	1.507
5	0.70	61.315	1.556
6	0.75	61.315	1.598
7	0.80	61.509	1.636
8	0.85	61.509	1.668
9	0.90	61.509	1.698
10	0.95	61.509	1.726
11	1.00	61.509	1.788

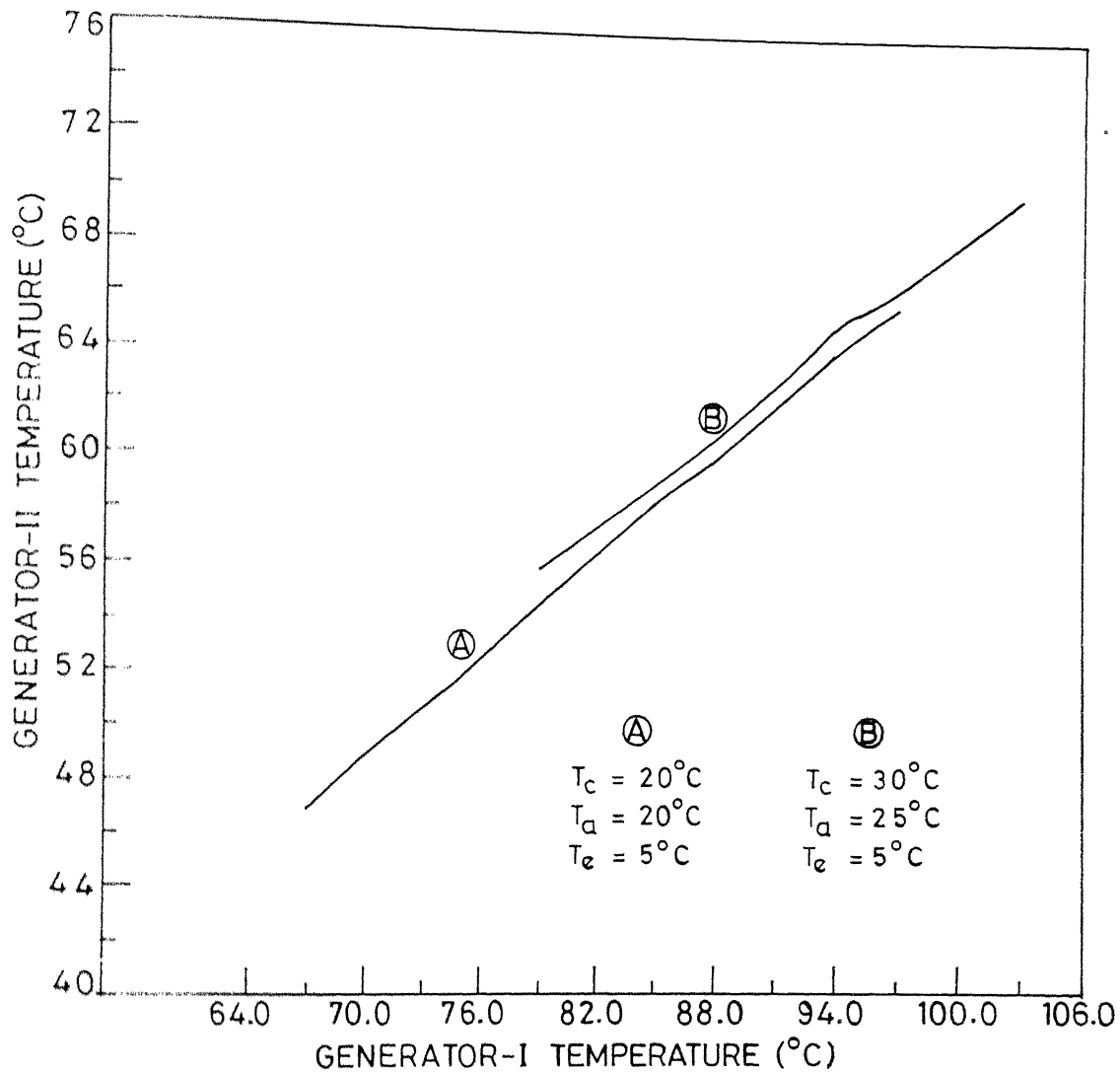


FIG. 5.9 VARIATION OF GENERATOR-II TEMPERATURE WITH GENERATOR-I TEMPERATURE .

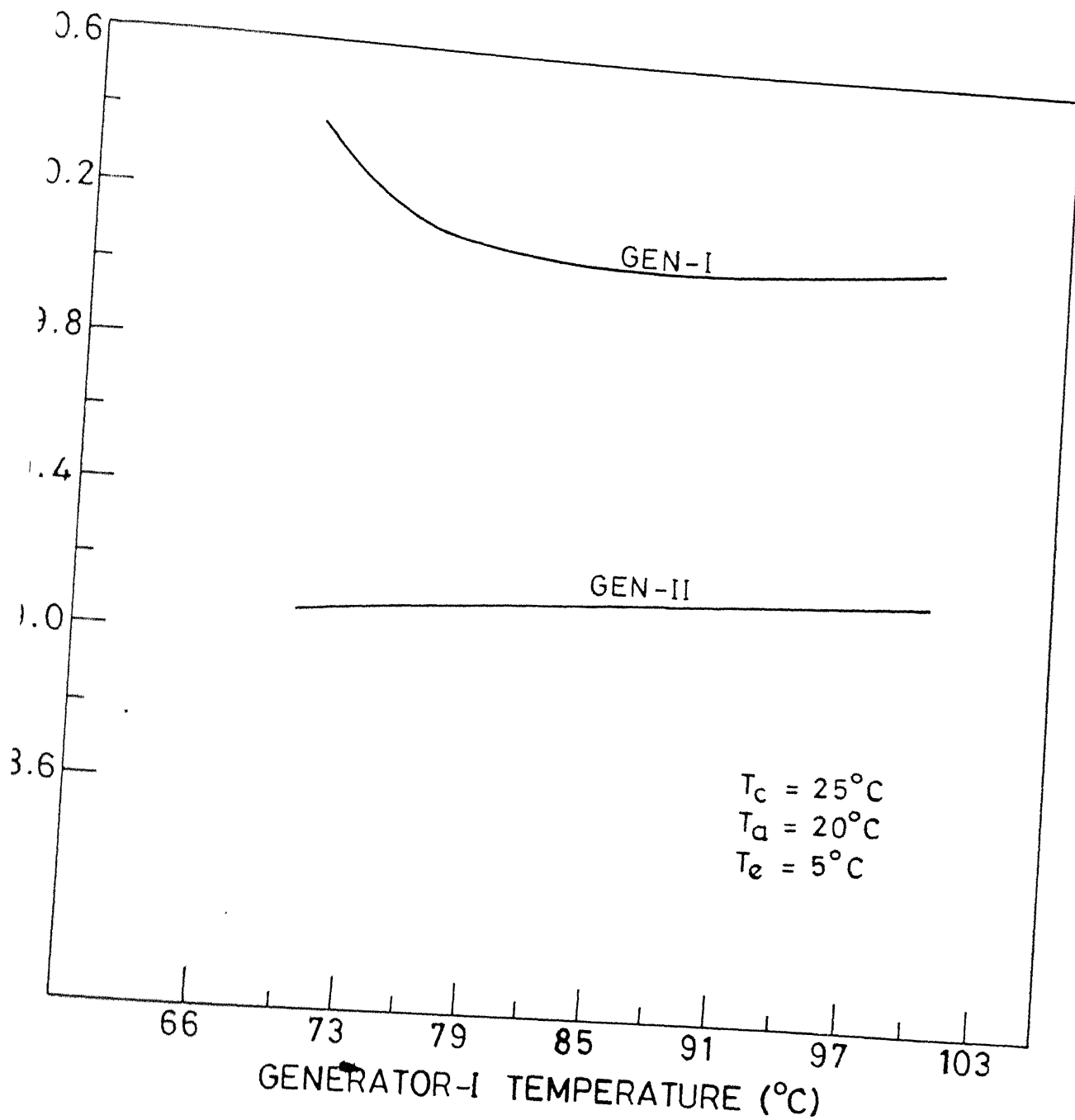


FIG.5.10 VARIATION OF GENERATOR\$ LOADSWITH GENERA-TOR-I TEMPERATURE.

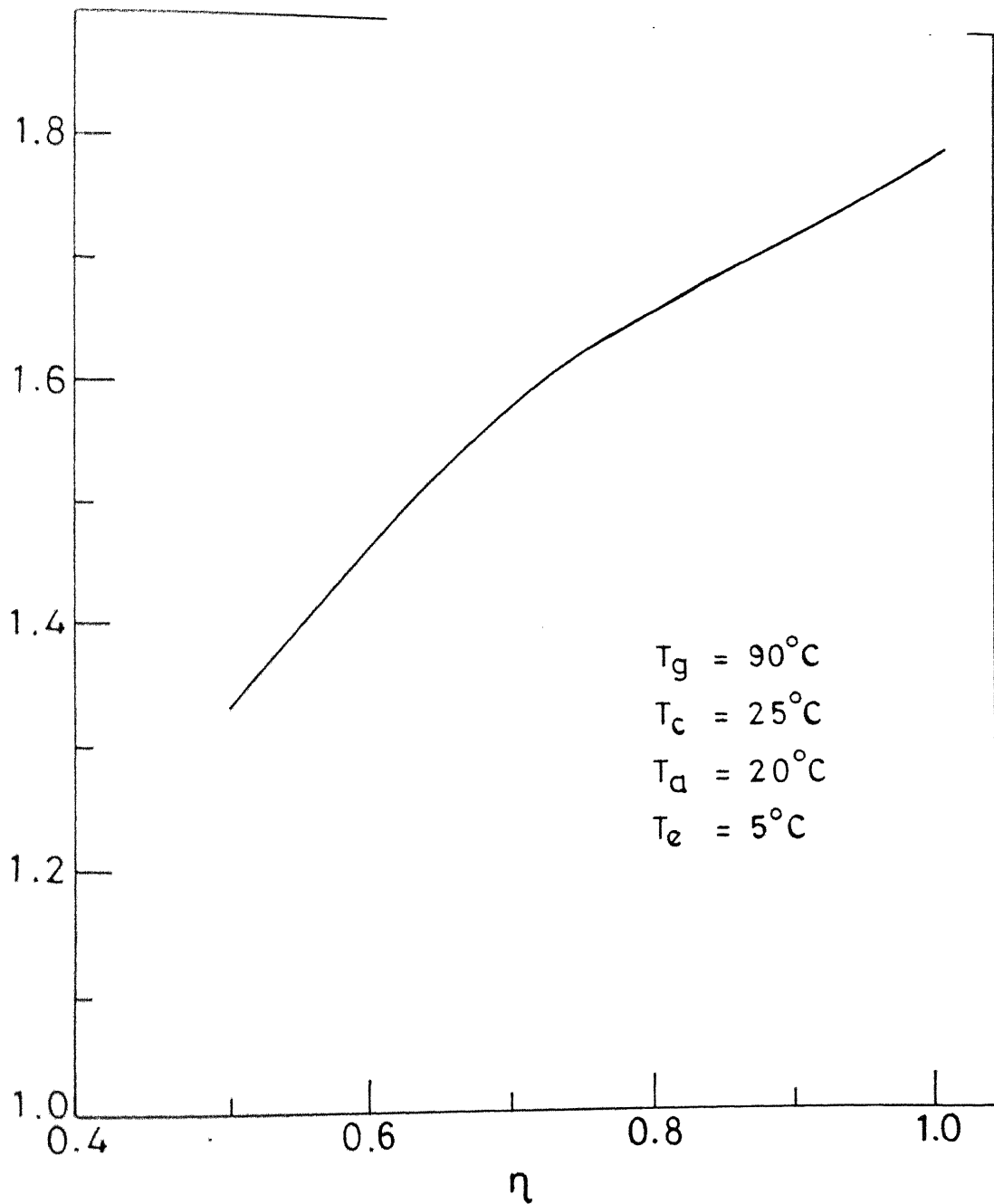


FIG.5.11 VARIATION OF C.O.P. WITH HEAT EXCHANGER EFFECTIVENESS.

CONCLUSIONS

An assessment based on steady state thermodynamic analysis of a double effect LiBr-H₂O absorption refrigeration cycle has been made, and its application to space cooling is considered as the main objective. A computer model of the system, based on the simultaneous solution of total mass, material and heat balance equations of different components in the system, has been developed. The proposed FORTRAN programme is very general in nature and simply by changing the input data, the required operational parameters can be determined quite easily.

In view of the results discussed earlier, the following conclusions may be drawn:

- (1) The C.O.P. obtained in a double stage-system is quite high (1.785 maximum) as compared to single stage (0.84 maximum). However, very high input source-temperature is required for system's operation. This criterion restricts its use to only those places, where very high-temperature steam, bio-gas, hot-water, waste-heat from flue gases and/or advanced solar collectors (evacuated tube), which could give reasonable collector efficiency at high operating

temperatures, are available. However, at locations where only/also high temperature heat sources are available and where the single-stage-system cannot work, a two-stage-system is the solution. Infact, in most of the process industries, the waste heat which is available at quite high temperature, may be successfully utilized for efficient space-cooling purposes.

(ii) This system has very high value\$ of C.O.P., almost approaching the ideal value at high condenser temperatures. The practicability of using high condenser-temperatures obviates the necessity of using a cooling tower along with the condenser and an air cooled condenser may be used for the purpose. This helps in a reduction in total cost of the system. Hence, the system would function more effectively in arid regions, where humidity remains low and the ambient temperature is high.

It may, thus, be concluded that a double-stage-vapour-absorption-system is more efficient than a vapour-compression-system in terms of the annual running cost as it does not require compressor for its operation, which consumes most of the energy required for the operation of a vapour-compression-system.

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APPENDIX -1State equations for aqueous Lithium-Bromide solution

The well known state equations, relating the solution-temperature and refrigerant-pressure to the refrigerant-temperature, the solution-enthalpy to the solution-temperature, enthalpy of superheated vapour of refrigerant to the evaporator and condenser-temperatures and the specific heat of the solution to the solution-concentration are given as following [26]:

$$t_f = A t'_f + B \quad (A-1)$$

$$\log_{10} P = C + \frac{D}{(t'_f + 459.72)} + \frac{E}{(t'_f + 459.72)^2} \quad (A-2)$$

$$H_f = p+q t_f + r t_f^2 \quad (A-3)$$

$$H_g = 1.925 T_g - 0.126 T_c + 2500 \quad (A-4)$$

$$C_{pL} = 4.259 - 0.053843 x + (2.307E-4) x^2 \quad (A-5)$$

where,

$$A = 2.00753 + 0.16976 x - (3.133362 E-3) x^2 \\ + (1.97668 E-5) x^3$$

$$B = 521.128 - 19.322 x + 0.374382 x^2 - (2.0637 E-3) x^3$$

$$C = 0.21147$$

$$C_{pL} = \text{Specific heat of LiBr-H}_2\text{O solution in kJ/kg.K}$$

$$D = -2886.373$$

$$E = -257259.46$$

$$H_f = \text{enthalpy of solution in Btu/lb}$$

$$H_g = \text{enthalpy of super heated vapour in kJ/kg}$$

$$I = -1012.07 + 79.5387 X - 2.358016 X^2 \\ + 0.03051583 X^3 - (1.400261 \text{ E-}4) X^4$$

$$J = 4.68108 - (3.037766\text{E-}1) X + (8.44845 \text{ E-}3) \\ - (1.047721 \text{ E-}4) X^3 + (4.80097 \text{ E-}7) X^4$$

$$K = -4.9107 \text{ E-}5 + (3.831841 \text{ E-}4) X - (1.07896 \text{ E-}5) \\ + (1.3152 \text{ E-}7) X^3 - (5.897 \text{ E-}10) X^4$$

$$t_f = \text{solution temperature (}^\circ\text{F)}$$

$$40 \leq t_f \leq 350 \text{ }^\circ\text{F}$$

$$t'_f = \text{refrigerant temperature (}^\circ\text{F)}$$

$$0 \leq t'_f \leq 230 \text{ }^\circ\text{F}$$

$$\frac{-2E}{10 + \sqrt{D^2 - 4E(C - \log_{10} P)} - 7^{0.5}}$$

$$X = \text{Percentage by mass LiBr in LiBr-H}_2\text{O solution}$$

$$45 \% \leq X \leq 70 \%$$

Conversion Factors -

$$(1) \quad 1 \text{ Btu} = 1.0155 \text{ kJ}$$

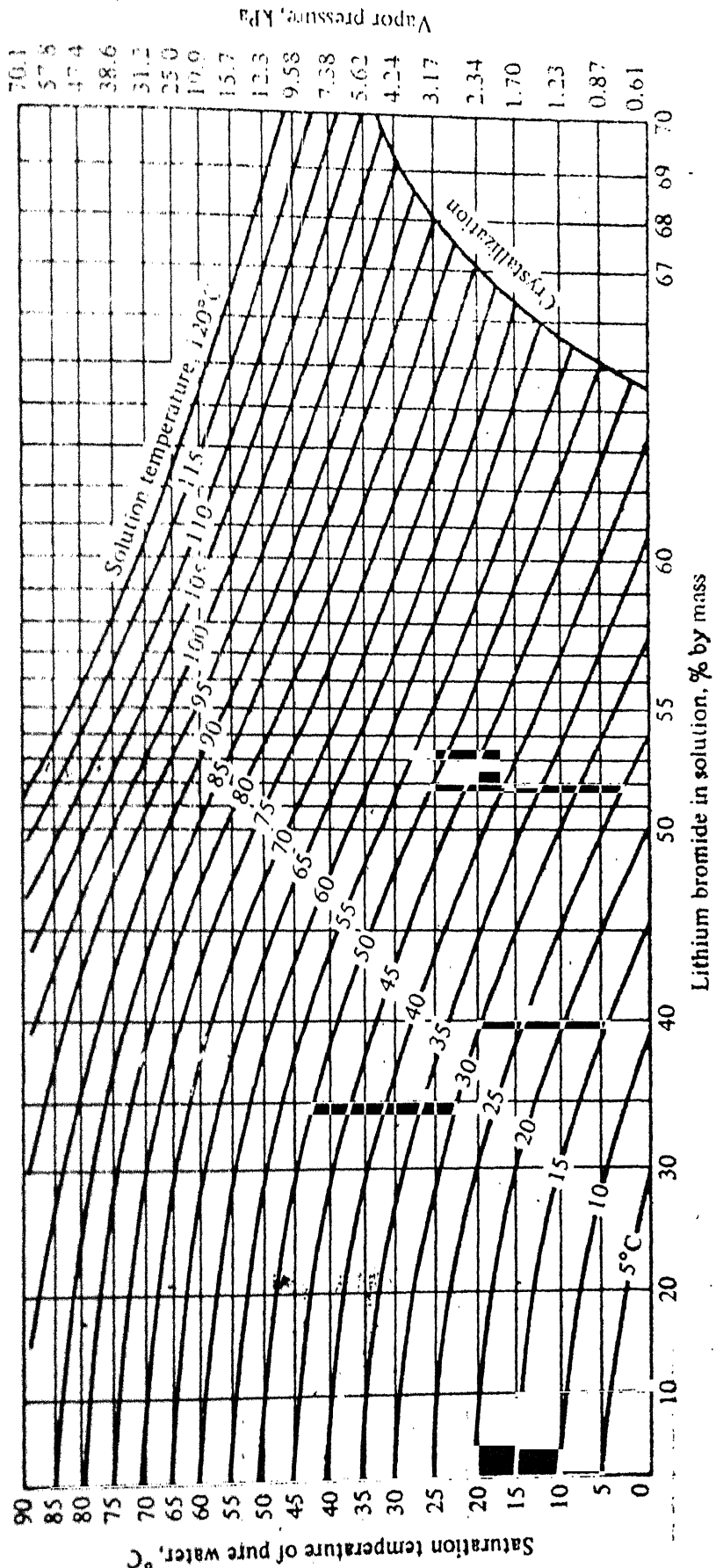
$$(2) \quad 1 \text{ Ton of Refrigeration} = 210 \text{ kJ/min}$$

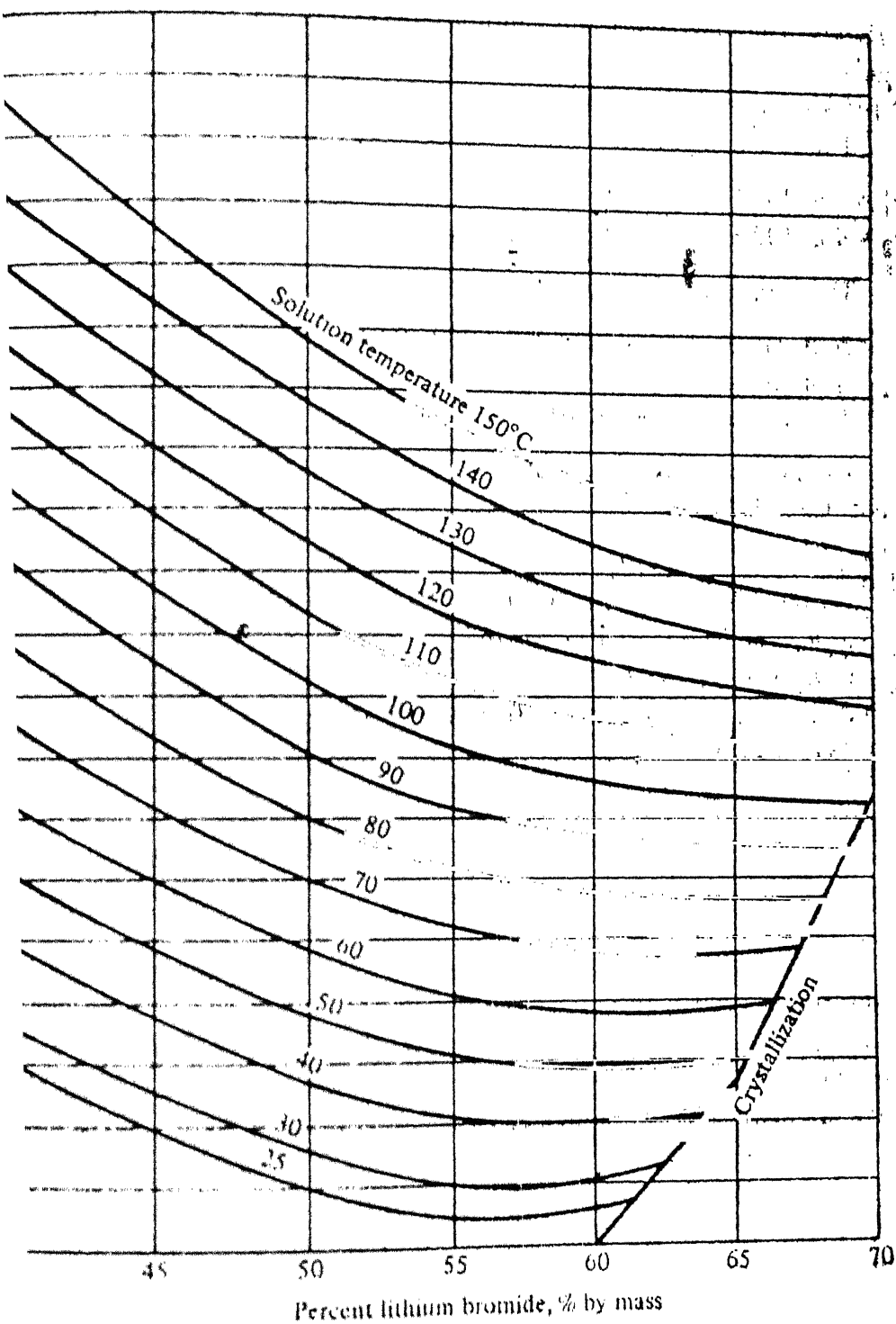
$$(3) \quad 1 \text{ Psia} = 6.9847 \text{ k Pa}$$

$$(4) \quad 1 \text{ Psia} = 52.3893 \text{ mm of Hg}$$

$$(5) \quad 1 \text{ Btu/lb} = 2.326 \text{ kJ/kg}$$

1. The following data were obtained from the above experiments:
 2. The solubility of lithium bromide in water at 20°C is 15.7 g/100 g water.
 3. The solubility of lithium bromide in water at 30°C is 25.0 g/100 g water.
 4. The solubility of lithium bromide in water at 40°C is 38.6 g/100 g water.
 5. The solubility of lithium bromide in water at 50°C is 57.8 g/100 g water.
 6. The solubility of lithium bromide in water at 60°C is 70.1 g/100 g water.
 7. The solubility of lithium bromide in water at 70°C is 85.0 g/100 g water.
 8. The solubility of lithium bromide in water at 80°C is 100.0 g/100 g water.
 9. The solubility of lithium bromide in water at 90°C is 115.0 g/100 g water.
 10. The solubility of lithium bromide in water at 100°C is 123.0 g/100 g water.
 11. The solubility of lithium bromide in water at 110°C is 138.0 g/100 g water.
 12. The solubility of lithium bromide in water at 120°C is 157.0 g/100 g water.
 13. The solubility of lithium bromide in water at 130°C is 170.0 g/100 g water.
 14. The solubility of lithium bromide in water at 140°C is 187.0 g/100 g water.
 15. The solubility of lithium bromide in water at 150°C is 201.0 g/100 g water.
 16. The solubility of lithium bromide in water at 160°C is 217.0 g/100 g water.
 17. The solubility of lithium bromide in water at 170°C is 234.0 g/100 g water.
 18. The solubility of lithium bromide in water at 180°C is 250.0 g/100 g water.
 19. The solubility of lithium bromide in water at 190°C is 267.0 g/100 g water.
 20. The solubility of lithium bromide in water at 200°C is 283.0 g/100 g water.
 21. The solubility of lithium bromide in water at 210°C is 299.0 g/100 g water.
 22. The solubility of lithium bromide in water at 220°C is 312.0 g/100 g water.
 23. The solubility of lithium bromide in water at 230°C is 323.0 g/100 g water.
 24. The solubility of lithium bromide in water at 240°C is 334.0 g/100 g water.
 25. The solubility of lithium bromide in water at 250°C is 344.0 g/100 g water.
 26. The solubility of lithium bromide in water at 260°C is 353.0 g/100 g water.
 27. The solubility of lithium bromide in water at 270°C is 361.0 g/100 g water.
 28. The solubility of lithium bromide in water at 280°C is 368.0 g/100 g water.
 29. The solubility of lithium bromide in water at 290°C is 374.0 g/100 g water.
 30. The solubility of lithium bromide in water at 300°C is 379.0 g/100 g water.
 31. The solubility of lithium bromide in water at 310°C is 383.0 g/100 g water.
 32. The solubility of lithium bromide in water at 320°C is 386.0 g/100 g water.
 33. The solubility of lithium bromide in water at 330°C is 388.0 g/100 g water.
 34. The solubility of lithium bromide in water at 340°C is 389.0 g/100 g water.
 35. The solubility of lithium bromide in water at 350°C is 390.0 g/100 g water.
 36. The solubility of lithium bromide in water at 360°C is 390.0 g/100 g water.
 37. The solubility of lithium bromide in water at 370°C is 390.0 g/100 g water.
 38. The solubility of lithium bromide in water at 380°C is 390.0 g/100 g water.
 39. The solubility of lithium bromide in water at 390°C is 390.0 g/100 g water.
 40. The solubility of lithium bromide in water at 400°C is 390.0 g/100 g water.
 41. The solubility of lithium bromide in water at 410°C is 390.0 g/100 g water.
 42. The solubility of lithium bromide in water at 420°C is 390.0 g/100 g water.
 43. The solubility of lithium bromide in water at 430°C is 390.0 g/100 g water.
 44. The solubility of lithium bromide in water at 440°C is 390.0 g/100 g water.
 45. The solubility of lithium bromide in water at 450°C is 390.0 g/100 g water.
 46. The solubility of lithium bromide in water at 460°C is 390.0 g/100 g water.
 47. The solubility of lithium bromide in water at 470°C is 390.0 g/100 g water.
 48. The solubility of lithium bromide in water at 480°C is 390.0 g/100 g water.
 49. The solubility of lithium bromide in water at 490°C is 390.0 g/100 g water.
 50. The solubility of lithium bromide in water at 500°C is 390.0 g/100 g water.
 51. The solubility of lithium bromide in water at 510°C is 390.0 g/100 g water.
 52. The solubility of lithium bromide in water at 520°C is 390.0 g/100 g water.
 53. The solubility of lithium bromide in water at 530°C is 390.0 g/100 g water.
 54. The solubility of lithium bromide in water at 540°C is 390.0 g/100 g water.
 55. The solubility of lithium bromide in water at 550°C is 390.0 g/100 g water.
 56. The solubility of lithium bromide in water at 560°C is 390.0 g/100 g water.
 57. The solubility of lithium bromide in water at 570°C is 390.0 g/100 g water.
 58. The solubility of lithium bromide in water at 580°C is 390.0 g/100 g water.
 59. The solubility of lithium bromide in water at 590°C is 390.0 g/100 g water.
 60. The solubility of lithium bromide in water at 600°C is 390.0 g/100 g water.
 61. The solubility of lithium bromide in water at 610°C is 390.0 g/100 g water.
 62. The solubility of lithium bromide in water at 620°C is 390.0 g/100 g water.
 63. The solubility of lithium bromide in water at 630°C is 390.0 g/100 g water.
 64. The solubility of lithium bromide in water at 640°C is 390.0 g/100 g water.
 65. The solubility of lithium bromide in water at 650°C is 390.0 g/100 g water.
 66. The solubility of lithium bromide in water at 660°C is 390.0 g/100 g water.
 67. The solubility of lithium bromide in water at 670°C is 390.0 g/100 g water.
 68. The solubility of lithium bromide in water at 680°C is 390.0 g/100 g water.
 69. The solubility of lithium bromide in water at 690°C is 390.0 g/100 g water.
 70. The solubility of lithium bromide in water at 700°C is 390.0 g/100 g water.
 71. The solubility of lithium bromide in water at 710°C is 390.0 g/100 g water.
 72. The solubility of lithium bromide in water at 720°C is 390.0 g/100 g water.
 73. The solubility of lithium bromide in water at 730°C is 390.0 g/100 g water.
 74. The solubility of lithium bromide in water at 740°C is 390.0 g/100 g water.
 75. The solubility of lithium bromide in water at 750°C is 390.0 g/100 g water.
 76. The solubility of lithium bromide in water at 760°C is 390.0 g/100 g water.
 77. The solubility of lithium bromide in water at 770°C is 390.0 g/100 g water.
 78. The solubility of lithium bromide in water at 780°C is 390.0 g/100 g water.
 79. The solubility of lithium bromide in water at 790°C is 390.0 g/100 g water.
 80. The solubility of lithium bromide in water at 800°C is 390.0 g/100 g water.
 81. The solubility of lithium bromide in water at 810°C is 390.0 g/100 g water.
 82. The solubility of lithium bromide in water at 820°C is 390.0 g/100 g water.
 83. The solubility of lithium bromide in water at 830°C is 390.0 g/100 g water.
 84. The solubility of lithium bromide in water at 840°C is 390.0 g/100 g water.
 85. The solubility of lithium bromide in water at 850°C is 390.0 g/100 g water.
 86. The solubility of lithium bromide in water at 860°C is 390.0 g/100 g water.
 87. The solubility of lithium bromide in water at 870°C is 390.0 g/100 g water.
 88. The solubility of lithium bromide in water at 880°C is 390.0 g/100 g water.
 89. The solubility of lithium bromide in water at 890°C is 390.0 g/100 g water.
 90. The solubility of lithium bromide in water at 900°C is 390.0 g/100 g water.
 91. The solubility of lithium bromide in water at 910°C is 390.0 g/100 g water.
 92. The solubility of lithium bromide in water at 920°C is 390.0 g/100 g water.
 93. The solubility of lithium bromide in water at 930°C is 390.0 g/100 g water.
 94. The solubility of lithium bromide in water at 940°C is 390.0 g/100 g water.
 95. The solubility of lithium bromide in water at 950°C is 390.0 g/100 g water.
 96. The solubility of lithium bromide in water at 960°C is 390.0 g/100 g water.
 97. The solubility of lithium bromide in water at 970°C is 390.0 g/100 g water.
 98. The solubility of lithium bromide in water at 980°C is 390.0 g/100 g water.
 99. The solubility of lithium bromide in water at 990°C is 390.0 g/100 g water.
 100. The solubility of lithium bromide in water at 1000°C is 390.0 g/100 g water.





APPENDIX - 6

Condenser Data for Vapour Compression System
(For Refrigerant-Freon 22), taken from the
Central Air Conditioning Unit, I.I.T. Kanpur.

(1)	Refrigerant Pressure	230 psig
(2)	Water Pressure	Inlet 40 psig Outlet 30 psig
(3)	Water temperature	Inlet 70°F Outlet 85°F
(4)	Pipe size of condenser	ID=5/8 dia(16mm) OD= 18 mm Length=10,126 tubes
(5)	Pipe material	Cu tubes, steel shell
(6)	No. of working hours	24
(7)	No. of condensing units	9
(8)	Compressors	7 Nos. -90 h.p. 2 Nos.-100 h.p.
(9)	Condenser Pumps	3 Nos.- 30 h.p. 1 No. - 25 h.p.
(10)	Full Load	720 tons
(11)	Operating Load (Month-May)	550 tons.

APPENDIX-7

Average Roughness of Commercial Pipes [18]

Material (New)	ϵ in mm
Riveted Steel	0.9-9.0
Concrete	0.3-3.0
Wood Stave	0.18-0.9
Cast Iron	0.26
Galvanized Iron	0.15
Commercial Steel	0.046
Drawn Tubing	0.0015

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